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MECHANICAL
ENGINEERING PRACTICE

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MECHANICAL ENGINEERING PRACTICE

A Laboratory Reference Text

BY

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AND

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THIRD EDITION
TENTH IMPRESSION

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MECHANICAL ENGINEERING PRACTICE

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PREFACE TO THE THIRD EDITION

In keeping with the original object of providing a self-contained textbook and handbook on experimental mechanical engineering, this edition features an increase in practical reference material. More tables, charts, and performance data have been included to assist the student in the interpretation of his data and results. Emphasis on fundamental instrumentation and test methods has been retained. But the entire text has been revised to obtain the best correlation and introduction of new material with ample cross references; and new test code requirements and recent instrument developments have been carefully included. A means for conserving student time is offered by the 100 suggested forms for tabulation of data and results.

In recognition of the value of simple but practical exercises for the beginning student, the usual work on instruments has been expanded to include such familiar items as automobile engine tune up, measurements by domestic gas and water meters, sampling and testing of fuels and lubricants, tests of belt drives, propeller fans, house-heating furnaces, boilers, radiators, and other common devices.

Many laboratory experiments have been included in the newer fields such as fluid mechanics, automotive and diesel engines, heat transfer, and air conditioning. Thus experimental and reference material is provided for advanced courses, as well as for a broader application of fundamentals.

The authors wish to thank the many engineering teachers who have made constructive suggestions based on their experience with this book, and to acknowledge indebtedness to the several engineering societies and firms who have permitted the reproduction of their published material.

C. F. S.
G. L. T.

MINNEAPOLIS, MINN.,
CLEVELAND, OHIO,
June, 1941.

PREFACE TO THE FIRST EDITION

A basic course in the theory and practice of heat power engineering is generally included as part of the training of the graduate engineer. This book is intended primarily as a text for use in the laboratory work of such courses in engineering schools. It may also be used as a reference book for the student after graduation. Advanced work in steam power plant testing has been included, as have the more important experiments which supplement the elementary courses in other divisions of mechanical engineering.

In the interest of conserving the time of the student, a definite outline of treatment has been used in the presentation of each laboratory problem. The authors feel that the average student entering the laboratory cannot be expected to proceed immediately on his own initiative and that requiring him to do so results in his bewilderment and a loss of his time. The treatment of the elementary experiments is, therefore, detailed and does not presuppose much engineering background, and the student is cautioned regarding the difficulties which he may encounter in the experiment. In the more advanced experiments the student is to a much greater extent thrown upon his own resources. Sufficient related material has been included in connection with each experiment, so that the better students may have opportunity to utilize the full time of the laboratory period to advantage.

The connection of the problem in hand to engineering practice has been indicated in the preface to each experiment. Information is furnished which will enable the student to compare his findings with typical commercial test results.

The interrelation of the various courses in the engineering curriculum is emphasized by frequent references, and by the drawing of material from the content of these courses. Although the inclusion of an ample number of reference tables and diagrams makes the book self-contained, the lists of references give the more advanced student ready access to both standard texts and recent technical papers in each field.

The test codes, transactions, and research publications of the several engineering societies have been freely used as source material, and although few direct quotations are made, the sources have been indicated. The authors wish to express their indebtedness to these societies and to their members, and also to the several engineering teachers who have offered criticisms and suggestions.

C. F. S.
G. L. T.

MINNEAPOLIS, MINN.,
LUBBOCK, TEXAS,
January, 1930.

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MECHANICAL ENGINEERING PRACTICE

A LABORATORY REFERENCE TEXT

CHAPTER I

INTRODUCTION

General Instructions to Students.—Keep the classroom theory in mind continually and note its application in practice. Do not overlook the fact that textbooks used in other courses contain excellent reference material.

Take much care in the preparation of your reports; your individuality should show clearly in your general presentation.

During your entire course, seek to familiarize yourself with the laboratory machines, their operation, care, and adjustments. Learn to handle instruments properly, inquire and, when possible, determine their precision, adaptation, limitations, etc.

Learn to work with the group so that your individuality is lost in the work of the group as a unit, but do not permit the other man to do your thinking or your work.

The experiments undertaken in the laboratory should be studied and carried forward with the same care and degree of accuracy and completeness as are required of the engineer in commercial practice.

The original notes taken in the laboratory should be carefully preserved and a copy handed in by the chief of the party. These notes should be taken neatly and should contain enough detail to enable an outsider to interpret them fully. Inexperienced observers are prone to use abbreviations or arbitrary symbols in their notes. The notes cannot then be interpreted by others, and even the observer himself is likely to forget their meaning. It is imperative that *all* original data as read from instruments or secured otherwise appear in the log or notes of the test.

Read the Preface to this manual.

All tools, instruments, apparatus, blank forms, reference books, catalogues, tables, etc., are to be secured on check from the instrument room, but these are not to be taken from the laboratory.

Precautions.—Carelessness is the cause of most accidents. It is the duty of every man to not only safeguard his own person but also that of others.

Do not under any condition start up or tamper with any machine, or open or close any valve unless you have been detailed to do so.

In all cases where the method of procedure is doubtful, do not blunder ahead and perhaps endanger yourself and others. Ask for specific instructions.

Always be on the alert for new sounds and noises and search out their cause.

If it is necessary to wipe or oil a moving part, use good judgment, keep eyes on your work. Do not wipe around gears with loose waste or work with unbuttoned sleeves or blouse.

Report all breakages to the instructor in charge or to the instrument-room caretaker.

Laboratory Reports—Preparation of the report is a definite and essential part of the laboratory experiment or problem. Punctuality in the submission of the report is a determining factor in the term grade.

> Next to ability in handling men, the success of an engineer probably depends more on his skill in writing clear, concise, and comprehensive reports than on any other factor.[1]

The engineering report is, in general, a product by which the engineer is judged. The ability to write a good report may be acquired through practice.

In laboratory reports on experimental or test work, the form of presentation below should in general be adopted. While this outline is not to be considered stereotyped, it does, however, represent a logical and convenient distribution of the material and will serve as a guide in the formation of practically any type of laboratory report.

In cases where the form and content of a report are not definitely indicated, the engineer is called upon to use his judgment regarding the same. Keep in mind that clearness demands **simple and accurate English** and logical order. It also calls for a judicious use of the tabular form, of line drawings or photographs, and of graphical presentation. Proper "display" is a great aid to clearness and readability in reports of all kinds. *Too many engineering reports are filed away practically unread because their writers paid too little attention to the display and arrangement of material presented.* This is especially true of reports submitted to busy executives. The combination of conciseness with comprehensiveness is greatly aided by the use of graphical representation.

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[1] Quoted from a publication of the U. S. Geological Survey.

To illustrate some of the possibilities, the assignments for reports covering the work in the mechanical engineering laboratory are varied. In some courses in particular, the available time is very short. In certain cases the experimental work occupies most of the available time. Unless the student receives special written or verbal instructions to the contrary, however, the rules for report writing as given here should be followed *in full.*

PLAN OF REPORT

1. **Title** of the experiment, brief but fully descriptive.

2. **Object** of the experiment, concisely stated (in past tense).

3. **Theory.**—Explanations of the principles or formulas involved and definitions of unfamiliar terms.

4. **Apparatus or Machine Tested.**—A brief description with such diagrams, sketches, blueprints, etc., as will make clear the general arrangement or setup. This section and the next are especially important where new apparatus or special devices are used.

5. **Instruments.**—List all instruments and auxiliary apparatus used, with laboratory identification numbers and concise description. The description may be omitted if already given in a previous experiment. Make such reference.

6. **Method.**—Describe method used, noting carefully any special precautions necessary in taking data or in securing proper functioning of the setup. Give duration of test and frequency of observations.

7. **Data.**—Copy all observed data, arranging the same in a neat and compact tabulation and, in general, observing the rule that calculated values should be preceded by the instrument

readings from which they have been determined. Sometimes, in the case of long tests during which voluminous data have been taken, a tabulation may be prepared on tracing cloth, and blueprints included in the report. In certain cases a "graphical log" may be prepared, or photographs of recording charts used to show the variations of the most important readings. The tabulations then usually contain only the averages for each run. Include sample indicator cards, graphic charts, etc., when taken, with full identification.

8. Results.—Arrange all results in tabular form. A well-planned tabulation of data and results is one of the most important parts of an engineering report. Give formulas used, with sample calculations. Headings 7 and 8 may very often to advantage be combined in one table.

9. Curves.—Plot curves in all cases where successive readings or results of dependent quantities enter. Plotted points are to be represented by $^3/_{32}$-in. circles. Each sheet should have a title and each curve should be labeled. Plot the curves to scales that present the characteristics with the greatest clearness. It is preferable to start the zero of both scales at the origin (located at lower left-hand corner of the sheet) if clearness is not sacrificed. Draw a **smooth** curve so that it best fits the plotted points. Discrepancies between curve and plotted points indicate errors in observations or computations, excepting in the cases of calibration curves and graphical logs that are inherently broken-line graphs. Curve sheets are to be done in India ink and curves inked in with a ruling pen. Before using a celluloid curve, however, it is well to sketch the curve freehand with a sharp pencil.

To differentiate, where a group of curves plotted to the same coordinates appear on the same sheet, the plotted points may be designated by open, filled or dotted circles, triangles, squares, or combinations of these symbols, and the curves represented by full, dotted, dot-dash, etc. lines. Sometimes it may be preferable to use colored inks.

10. Discussion.—The result of your own thinking is desired under this heading. The conclusions arrived at are of prime importance. *The entire project was undertaken in order to reach definite conclusions.*

State what your results prove, keeping in mind the object and referring definitely to your curves and tabulations.

Designate by blocking out in red ink, underlining, caps, etc., the most important results obtained.

Give reasons for results differing from theory or standard practice. Give value and quote authority in making comparisons.

Make a careful analysis of the probable errors in your experiment or test work and what precautions you would take to minimize such errors if you were to repeat the experiment.

Give **constructive** criticism as to the method of solution employed, condition of apparatus, its dependability and accuracy.

11. Mechanical Details of Report.—Use standard student report paper, size 8½ by 11 in.

Bind each report as specified by the instructor.

Leave a 1½-in. margin at the left of the sheet for binding, a 1½-in. space or more at top of each sheet and a space of 1 in. or more at the bottom. Dimensions for margins should rest largely with the individual; endeavor always to present a well-balanced page.

Sketches and diagrams should be made to scale, each should have a title and all parts should be labeled for reference.

In general, avoid the use of abbreviations in the text of the report.

Spelling, punctuation, paragraphing, and other mechanical details of correctly written English are of utmost importance.

Number all sheets in the upper right-hand corner.

Use one side of the sheet only.

Do not attempt to make your first copy a finished report. Use a lead pencil and plenty of paper in working up your report from the outline form here presented. The final copy should be typewritten (double spaced) if possible, but pen-and-ink copies will be accepted.

Use of Tools and Care of Machinery.

Use each tool for the purpose for which it has been made.

Do not use wrenches or tools other than hammers as hammers.

Do not use screw drivers as chisels or levers.

Do not use pipe wrenches or pliers on square or hexagonal nuts or cap screws, or on piston rods, valve rods, or for purposes other than making up pipe joints.

Do not use a chisel and a hammer to loosen or tighten up a nut or cap screw.

Do not use emery paper for cleaning or brightening parts or for grinding flat valves or valve seats. Secure the proper material for this work.

When unscrewing stud bolts, nuts, pipes, or any parts that are threaded, do not apply force until you have determined whether the thread is right- or left-handed.

For driving out pins, do not use a center punch, but make use of a pin punch or straight rod.

Before taking down a machine, determine first what parts should be taken down in section and then the sections to be further dismantled.

If the parts of a machine refuse to separate, do not apply undue force, for this may cause breakage; search for the trouble. If a hammer is needed to separate the parts, make use of a wood block as a buffer.

Be careful not to score or dent any parts of a machine by the dropping of tools or otherwise. Be particularly careful of such parts as pistons, piston rods, valve seats, or any other parts that require a running fit or a smooth surface for perfect operation.

When taking down a machine, lay out the different parts in the same order in which they are taken down and when possible replace nuts, bolts, screws, etc., immediately. This will aid materially in the perfect fitting of parts when asembling, as well as expediting the erection.

Do not lay tools or parts in openings or spaces in the machine under examination. They may be covered up or shut in when assembling. When through using a tool, put it in your kit: do not lay tools about promiscuously.

CHAPTER II

MECHANICAL MEASUREMENTS—METHODS AND INSTRUMENTS

LABORATORY MEASUREMENTS

General Considerations regarding Measurements.—The instruments and measurements used by the mechanical engineer are essentially mechanical in nature, and most of them involve two common mechanical properties: force and displacement. The forces involved may be due to gravity, inertia, friction, elastic effect, magnetism, or centrifugal action. In many cases the force is resisted by a spring, and the deflection of the spring is the quantity actually measured. In addition to the force-and-displacement instruments there are others that depend primarily on linear or volumetric displacement only. The inaccuracies of mechanical instruments are largely due to mechanical imperfections and to accompanying mechanical difficulties, such as friction, inertia, vibration, looseness, and inaccuracy of linear scales.

Accuracy of Measurements.—Mechanical laboratory investigations are based fundamentally upon the art of measurement. The observations required for a given setup are usually of varying quantities and, therefore, subject to errors. Test codes issued by the engineering societies, in addition to providing methods to be followed in the making of tests, also call attention to known sources of error and to those precautions that can be taken to minimize their effects.

A condensed treatment of the errors (inherent or otherwise) entering the laboratory problem is therefore required in order to establish a rational method of approach to the fertile field of experimentation. Repeated observations of the same quantity, even when made with the same instrument and apparently under the same circumstances, will differ in value. Thus, every observation is subject to an error, the **error** being the difference between the observed value and the true value. When the observed value exceeds the true value the error is considered positive, and when less than the true value the error is considered negative. The possible errors entering an observed value may be classified as follows:

1. Mistakes or blunders.
2. Errors of method.
3. Inaccuracies in instruments.
4. Personal errors.
5. Accidental errors of observation.

In the laboratory the engineer attempts to arrive at the facts through observed experimental data. His effort as an experimenter should be to reduce all the above types of errors to a minimum. It will be understood, however, that to eliminate completely all the listed errors and arrive at absolute values is quite impossible. Suggestions for the avoidance of the common errors in the experiments presented in this book, are given in the Notes and Precautions following each experiment.

1. *Mistakes or Blunders.*—These errors are due primarily to carelessness on the part of the observer. Their magnitude is relatively large, and they follow no definite law as to regularity of occurrence. Errors of this nature are subject to control and complete elimination.

2. *Errors of Method.*—Observations that deviate from real values may in part be due to the method of application of the instruments in securing the data. Errors in method are involved in the characteristics of the quantity being measured. The avoidance of such errors is largely a matter of judgment and experience on the part of the experimenter. The characteristics of the quantity to be measured have much to do with the method to be used. Repetition of given measurements by the use of totally different methods may serve to check the presence of original errors. An agreement presumes that no fundamental error has been committed.

3. *Inaccuracies in Instruments.*—Errors due to physical sources are always present in instruments, such as, for instance, improper design, faulty construction, or imperfection of parts. The experimenter should have a general knowledge of the instrument field to enable him to select the proper instrument for a specific purpose. He should further acquaint himself fully with the inherent characteristics of the instrument selected, with a study of the proper placement of the instrument relative to other apparatus or equipment in the setup for test.

Accuracy in instruments may be determined by calibration (see Instrument Characteristics, page 9). The conditions during calibration should correspond closely to the conditions prevailing when the instrument was under test. For certain cases, it is very desirable to have the instruments calibrated in test position and under test conditions.

In evaluating the accuracy of an instrument, one of the most important factors is the length of the scale or the actual linear displacement of the indicating device. The scales on most indicating instruments are between 2½ and 6 in. in length, though large dial or disk-type scales may be longer. Round-style recording charts have scales from 2 to 5 in. long, but strip charts may have maximum scale lengths of 10 or 11 in. Instrument and chart graduations closer than 20 to the inch become difficult to read, and 10 to the inch is more desirable. Pen-record lines on recording charts may be $\frac{1}{32}$ in. or wider, but an edgewise pointer with mirror scale can be read more closely. With these physical limitations in mind, it is evident that an accuracy of 1 or 2 per cent of full scale is the best that can be expected with many common instruments. To determine the difference between two readings 1 in. apart on an instrument scale, within an accuracy of 1 per cent, calls for readings accurate to a linear distance of $\frac{1}{200}$ in., a near impossibility without a magnifying lens.

Relative accuracy of observations is usually much higher than absolute accuracy. Many instruments and methods are quite satisfactory for measuring *differences* in temperature, velocity, rate of flow, etc., but are not suitable for determining absolute values.

4. *Personal Errors.*—The experimenter, owing to his personal characteristics, traits, tendencies, etc., combined perhaps with a lack of experience, general training, and knowledge of laboratory technique, may introduce errors of considerable magnitude into his observations—errors in making interpolations, tendency to read high or low, inability to read rapidly, introducing time lag, etc.

The very nature of these errors presupposes that the tendencies in one direction are the same as in the opposite direction, *i.e.*, for instance, he is just as likely to read low as high and vice versa. In an extended series of observations, therefore, a balance would be established, as the positive and negative values would annul each other and the error would tend to cancel out.

5. *Accidental Errors of Observation.*—Measurements of the same quantity by the same observer with the same instruments and under the same conditions will invariably differ in value. The reason these discrepancies are introduced can in no case be assigned.

All the first four enumerated sources of error may be eliminated or correction determined and applied. If such correction be made, the reduced observation will nevertheless show a discrepancy or departure from the true value; such is termed the accidental error. Even when all the controllable conditions are kept as near constant as possible, accidental variations may appear at any moment and continue for any length of time, such as, for instance, the sticking of an indicator piston or increased friction of valves or piston of a steam engine due to a temporary lack of oil. It is not possible to determine the magnitude of the accidental error but only the probable limits between which the true result lies. The treatment of accidental errors by the laws of probability and the theory of least squares may be found in any text on least squares or in a manual of physical measurements.

Instrument Characteristics.—Certain characteristics are common to all instruments. A good discussion of the General

Considerations applying to mechanical engineering instruments is given under this heading in Part 1 of the A.S.M.E. Code on Instruments and Apparatus. The following is a summary.

Sluggishness is the tendency to respond slowly, or the lagging in an instrument, usually considered the opposite of sensitiveness. Sluggishness is normally measured by the amount of displacing effort required to start useful motion of the indicating element after the instrument has been at rest. The tested points should cover the full range of the instrument. A plotted curve would enable visualization of the variation through the entire range. The degree of sluggishness may be changed by tapping or jarring the instrument while observations are being made. Rapid variations may change the degree of sluggishness.

Sensitiveness is the property to discern small variations, usually due to extreme delicacy of adjustment in an instrument. Sensitiveness in an instrument is a very important property. It is a measure of the degree of refinement possible in use. Sensitiveness in excess of requirements as set by the accuracy desired in a given set of measurements is generally a detriment because it involves delicacy, slowness of operation, or some other undesirable quality. The numerical value of the sensitiveness is a characteristic dependent upon the initial design of the instrument.

The value of the sensitivity can be determined readily from the results of the calibration by noting the amount of the displacement occurring for given variations in the values of the quantities being measured. The displacement can then be determined per unit of value and the readable variation that follows from the design of the instrument.

The A.S.M.E. Code states that it is always necessary to compare the values of sensitivity determined as above with the value determined for sluggishness and variance. Sensitivity within the limits set by sluggishness and variance is useless, and an attempt to use it results only in wasted effort and, possibly, mistaken accuracy.

Accuracy, the freedom from error, is a purely relative consideration, since there is no such thing as absolute accuracy in an instrument. The degree of accuracy is dependent upon the condition of use, presence of vibration, rapidity and magnitude of variations of the measured quantities, etc. Accuracy may be determined through *calibration.*

The exact procedure to be followed in calibration varies with the type of instrument, use in service, and the desired degree of accuracy required. In general, the calibration indicates:

1. Sluggishness of instrument in relation to its sensitiveness.
2. Variance in its relation to sensitiveness. Variance represents the amount successive readings of the same quantity differ.

The calibration may be performed to determine the departure from absolute accuracy or the consistency of readings within the instrument itself. To determine the departure from absolute accuracy, the calibration must be made against a standard instrument or known values. If the calibration proceeds for both increasing and decreasing increments covering the full range of the instrument, then the two calibrations up and down plotted separately against true values will form a "hysteresis loop." Of course, for an absolute instrument (if such were possible to obtain), the plot would show a straight line, *i.e.*, the loop would squeeze into a straight line. The more accurate the instrument the narrower will be the loop.

For convenience in engineering work, it is customary to draw a single line to represent an average for the loop and use this as the calibration curve. Ordinarily, this curve is plotted directly from averages determined from the numerical values of the observations, and the separate calibrations showing the loop are not plotted. Actually, the calibration curve is not a line but rather a band of lines of definite width.

Laboratory Practice.—If any point of a plotted series lies markedly off the mean calibration line, it is evidence of an abnormal error, and the experimenter is justified in discarding the observation.

Curves aid in visualizing instrumental variations and are much easier to interpret and apply than are the corresponding columns of figures.

As a rule, instruments should be read to a fraction (preferably the nearest tenth) of their smallest division. The last figure in the reading is an estimated quantity but should be included in the result as the last significant figure. Quantities that fluctuate should be measured at regular time intervals and the true average determined for the period.

The instrument selected must have such sensitivity that the values of the quantities being measured can be read upon the instrument to at least as small a subdivision as is required by the condition of the test.

Aside from using an instrument of finer graduations the greatest aid to accuracy is a repetition of readings; especially is this required where the magnitude of the observed quantities is continuously changing. The more rapid the variation the greater the difficulty in measuring and, in general, the lower the accuracy attainable.

The length of test should be such as to permit the reduction of the error to the desired degree of accuracy. The A.S.M.E. Test Codes usually specify the minimum length of run for the various classes of machinery.

Preliminary computations of results from either a trial run immediately preceding the test or from the data secured at the beginning of the test will serve to check the correctness of the setup and the proper operation of the machinery and equipment. A progressive curve plotted as the test proceeds will show tendencies and furnish a continuous check on the validity of the data.

PRESSURE MEASUREMENTS

Experiments 1, 2, and 3

I. PREFACE

Pressure measurements are involved in a wide variety of engineering problems. When any reference is made to pressure in engineering work, the terms employed by the speaker or writer will depend largely on what instruments he has in mind for measuring the pressure and upon the manner in which these instruments are to be used. The power-plant engineer refers to "a steam pressure of 200 lb," or a "chimney draft of 1 in. of water." The hydraulic engineer speaks of a turbine operating with a "head of 60 ft" or of the "pressure drop" in a pipe line. The structural engineer takes into account the "wind pressure" on the wall of a building. The ventilating engineer speaks of a "static pressure of 2 in. of water and a velocity pressure of ½ in." at the outlet of a fan.

Since **pressure is defined as force per unit area,** it is apparent that measurements of pressure and of force are virtually the same thing. Measurements of weight, or the force of gravity, are discussed under a separate heading, page 53. Measurements of pressure are accomplished by balancing the pressure to be measured against a known pressure or force. In barometers, manometers, and dead-weight scales the pressure is balanced by lifting a weight, and the known force is that of gravity. In Bourdon and diaphragm gages the known force is produced by the deflection of a calibrated spring.

In most engineering pressure measurements, the unit of area is taken as 1 sq in. and the pressure is denoted in pounds total acting on this unit area. However, when the value is to be used in a formula (in thermodynamics or hydraulics), it often becomes necessary to use pounds per square *foot* in order to obtain the answer in the desired units.

Barometric or atmospheric pressure (usually read in inches of mercury) is the pressure, above the zero pressure or from absolute vacuum, corresponding to the pressure of the atmosphere. Barometric pressure may be expressed in pounds per square inch or in any other desired units.

Gage pressure is usually expressed in pounds per square inch and denotes pressure measured above atmospheric or barometric pressure, *i.e.*, the gage reading at atmospheric pressure is zero.

Absolute pressure is the pressure measurement above absolute zero pressure. If its value is greater than the barometric pressure it will be equal to the barometric pressure plus the gage pressure. If its value is less than the barometric pressure it will be equal to the barometric pressure minus the "vacuum" (see Fig. 1).

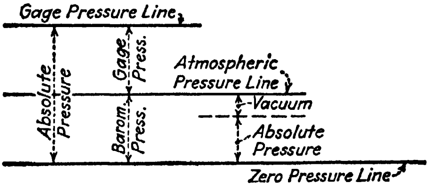

FIG. 1.—Kinds of static pressure.

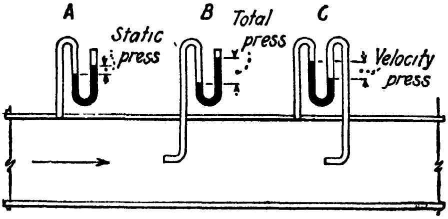

FIG. 2.—Measurement of static, total, and velocity pressure.

If a fluid is flowing through the pipe shown in Fig. 2, in the direction indicated by the arrow, and a pressure gage is connected as shown at *A*, this gage will indicate the internal or bursting pressure. Since this gage reading is not affected by the velocity of flow, this pressure is called by engineers the **static pressure.** If the gage is connected at *B* (Fig. 2) it will indicate, *in addition* to the static or bursting pressure, the pressure created by the impact of the flowing fluid. This combined pressure is termed the **total** or **kinetic pressure.** It is evident that the difference between the readings at *A* and *B* for any given rate of flow will give directly the pressure due to impact. This difference is termed the **velocity pressure,** and it is usually measured by the use of a "differential manometer," as at *C* of Fig. 2.

From the "velocity pressure," the actual velocity at which the fluid is moving in the pipe may be calculated. The velocity-pressure measurement, therefore, forms the basis for several types of metering devices for the measurement of fluid flow (see Chap. VII, page 282). When the two connections *A* and *B*, of Fig. 2, are combined into a single instrument, as shown in *E*, of Fig. 161 (page 281), the device is termed a "pitot tube." Whether in combination or not, the static tube is likely to give results of doubtful accuracy unless certain precautions are observed in its construction. Since this difficulty is most frequently encountered in connection with problems of fluid measurement, it is discussed in Chap. VII, page 273.

Friction pressure is created by the obstruction to flow offered by the pipe, fittings, valves, and other resistances. The friction pressure between any two points in a pipe line of constant cross section corresponds to the measurable "loss" in static pressure between the two points. Friction pressure therefore increases the static pressure required to produce a given flow by an amount equal to the loss caused by the resistances.

The words **pressure** and **head** are often used interchangeably. The weight of a column of fluid of unit cross section (creating a given pressure at its base), and its corresponding height, are mutually convertible through the use of a transformation factor. For accurate work, the temperature of the fluid must be taken into account. This is especially true when dealing with gases (in which case the increase in volume is directly proportional to the absolute temperature). The density of a gas for any pressure and temperature may be calculated from the equation

$$PV = wRT. \tag{1}$$

For values of R, see Table XIX, page 233.

II. INSTRUMENTS FOR PRESSURE MEASUREMENT

Manometers.—Manometers are the simplest form of pressure-measuring instruments. The difference of pressure on the two legs of the **open manometer** (Fig. 3) is indicated by the height h, and this indication gives gage pressure or vacuum, *i.e.*, the pressure above or below atmosphere. If it is desired to convert this reading to absolute pressure in pounds per square inch, the density of the fluids in the manometer and in the barometer must be taken into account (see Table I).

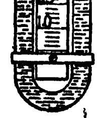

FIG. 3.—Simple open manometer.

The **manometer liquids** most commonly used are mercury, water, oil, and alcohol. Table I gives the densities of these liquids at 32 and at 70°F and also the conversion factors to change manometer readings into pressure in pounds per square inch.

Where greater accuracy is desired than that given by the vertical U-tube, an inclined tube may be used as represented in Fig. 4. In this particular form, it is usually called a **draft gage.**

Corresponding to the vertical rise *h*, the liquid in the tube moves through a straight-line distance *h′*, which is longer than *h* in the proportion of *h* csc α to *h*. The smaller the angle, the greater the length *h′*. This increase in length permits of finer scale graduations.

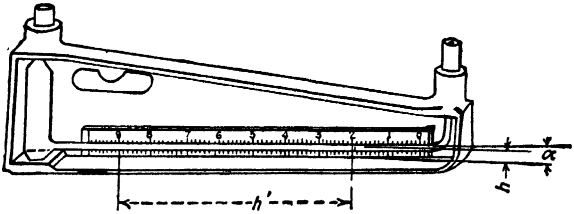

Fig. 4.—Inclined-draft gage.

The usual multiplying factors are from 2 to 1 to 10 to 1, but a multiplication of 20 to 1 can be obtained if the instrument is carefully used. The most common errors in the readings of inclined manometers are due to faulty setting of the zero of the scale or to inaccurate leveling of the instrument.

Table I.—Properties of Manometer Liquids

Manometer liquids	Specific gravity relative to water at 39 deg. fahr.*		To convert in. of liquid to lb. per sq. in. multiply by	
	Manometer temp. 70 deg. fahr.	Manometer temp. 32 deg. fahr.	Manometer temp. 70 deg. fahr.	Manometer temp. 32 deg. fahr.
Mercury	13.543	13.595	0.489	0.491
Water	0.9980	......	0.0361	......
Kerosene (44 to 46 deg. A.P.I.)	0.799	0.812	0.0289	0.0293
Red draft gage oil (approx.)	0.820	0.835	0.0296	0.0301
Methyl (wood) alcohol	0.790	0.810	0.0285	0.0293
Ethyl (grain) alcohol	0.789	0.805	0.0285	0.0291
Gasoline (58 to 60 deg. A.P.I.)	0.739	0.752	0.0267	0.0270

* Density of water at 39°F, 62.43 lb per cu ft.

If a large supply chamber or well is used as one leg of the manometer, as shown in Fig. 5, the resulting instrument is somewhat easier to read than is an ordinary U-tube. The zero of the scale is set at the liquid level in the supply chamber, and only one scale reading is then necessary.

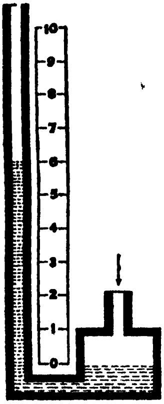

FIG. 5.—Cistern-type manometer.

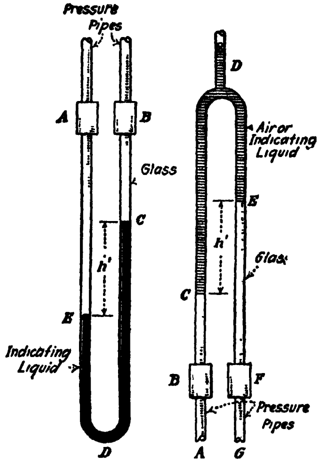

FIG. 6.—Upright and inverted U-tube manometers. (*A.S.M.E. Code.*)

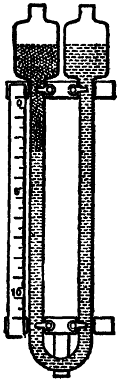

FIG. 7.—Two-fluid manometer.

The manometer, either vertical or inclined, is essentially a "differential" instrument, *i.e.*, it measures the difference between two pressures, and one of these may or may not be the atmospheric pressure. Figure 6 shows two ways of using a differential U-tube manometer for measuring the pressure drop across a metering orifice in a water pipe line. The manometer at *A* is filled with mercury and is located below the pipe, the connecting tubes being full of water. The manometer at *B* is inverted, and the water rises from the pipe into each tube. The heights of both indicating columns are affected equally by the amount of air trapped in (or pumped into) the top of the inverted U-tube.

A more sensitive gage than the standard U-tube may be obtained by using two fluids of different specific gravity such as oil and water (Fig. 7). Another type of two-fluid manometer using oil and mercury is shown in Fig. 8. Pressure introduced into leg *A* containing only mercury will raise the mercury in leg *B*. This rise of mercury will cause a greater rise of the oil above the mercury where the leg becomes much smaller in diameter. The pressure to be measured is counterbalanced partly by mercury

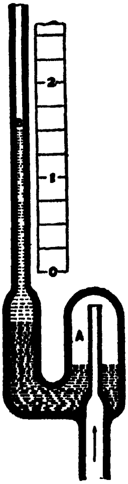

FIG. 8.—Two-fluid manometer, cistern type.

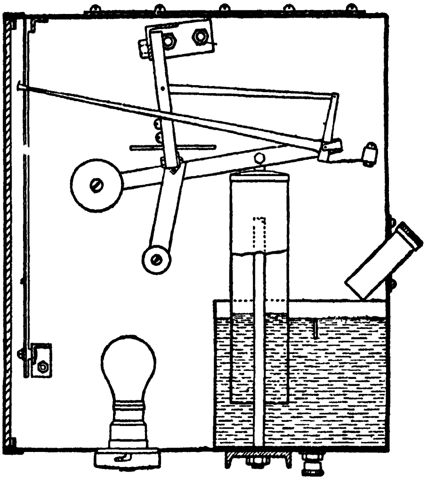

FIG. 9.—Inverted-bell manometer used on pointer-type draft gage. (*Ellison.*)

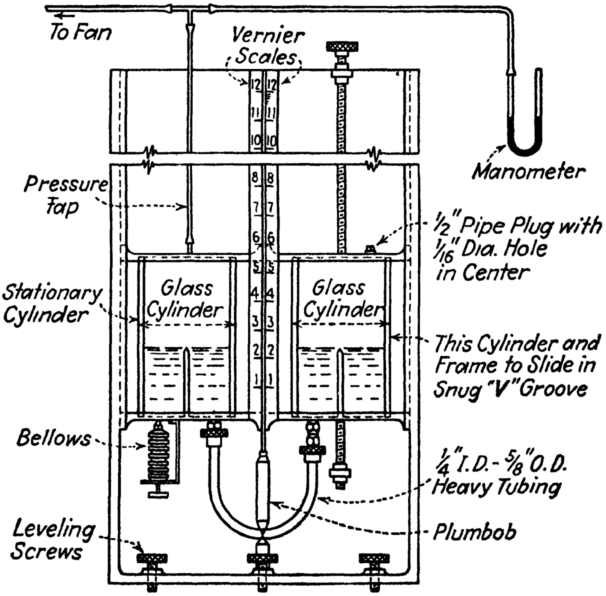

FIG. 10.—Hook-gage micromanometer. (*A.S.H.V.E. Fan Test Code.*)

and partly by oil. In this manner a scale reading is obtained which is much greater than if mercury alone were used.

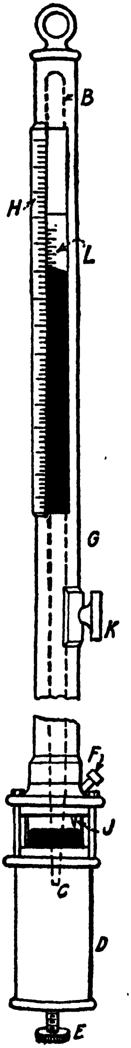

Fig. 11.—Commercial barometer.

Inverted bell manometers are widely used for permanently mounted draft gages, as shown in Fig. 9. They operate like a gasometer storage tank. Functionally, they are modified U-tube manometers, the inside of the bell acting as one leg of the manometer. Vertical displacement of the bell is balanced by a weight, and the displacement may be greatly magnified by a lever mechanism and a long pointer. Thus a large easy-reading scale may be used.

Micromanometers are manometers fitted with some precision reading or magnifying device. They are usually graduated in increments of 0.001 in. of water, though they may in some cases be sensitive to 0.0001 in. Most micromanometers are of the inclined-tube type. The **Wahlen gage** is a two-fluid micromanometer.

A **Hook gage** is a means for accurately measuring a liquid level, and it is sometimes applied to manometers, in order to obtain more accurate readings than are possible by visual observation of a meniscus. The essential feature of a hook gage is the piercing of the liquid surface by a sharp-pointed hook, raised from below. Just before the point pierces the skin of the liquid surface a pimple is seen to rise above the point. The point is lowered until this pimple is barely discernible and the position of the point, *i.e.*, the liquid level, is then read by means of a vernier or a micrometer screw. The Standard Fan Test Code of the A.S.H.V.E. requires that inclined manometers used in fan tests be calibrated in place against a water-filled hook gage reading to 0.001 in. of water (see Fig. 10).

Flowmeter Manometers are among the most important of the industrial pressure-measuring instruments. These are usually simple manometers enclosed in special cases with indicating, recording, and integrating devices. These latter devices are actuated by the level of the liquid in the manometer (usually mercury) using floats and lever arrangements, or by means of electrical or magnetic devices. It is thus possible to make the manometers distant reading.

The Barometer.—In the commercial Fortin type of barometer[1] (Fig. 11) a vernier *L*, indicating the height of mercury, is attached at the upper or reading end of the barometer. For ordinary purposes, this scale need be only 3 in. in length to cover the usual range of atmospheric fluctuations. The cistern of the commercial barometer is provided with a flexible leather bottom, which permits of adjustment of the mercury surface (by means of a screw piston *E*) so as to bring it to a standard level as indicated by a stationary needle point *J*. In making this adjustment, the mercury surface is brought to a point contact with the needle. To read the barometer, the vernier is adjusted so that the top of the meniscus is in line with the lower edge of the vernier slide.

TABLE II.—BAROMETER CORRECTIONS FOR TEMPERATURE

To correct the observed reading of a mercury barometer or U-tube to the 32° standard, add or subtract the following values in inches of mercury

Actual temperature of mercury column, °F	Observed reading of mercury column, in. Hg						
	20	22	24	26	28	30	32
	CORRECTIONS IN INCHES OF MERCURY Add the correction						
−20	0.09	0.10	0.11	0.11	0.12	0.13	0.14
0	0.05	0.06	0.06	0.07	0.07	0.08	0.08
20	0.02	0.02	0.02	0.02	0.02	0.02	0.02
	Subtract the correction						
40	0.02	0.02	0.02	0.03	0.03	0.03	0.03
60	0.06	0.06	0.07	0.07	0.08	0.08	0.09
80	0.09	0.10	0.11	0.12	0.13	0.14	0.15
100	0.13	0.14	0.15	0.17	0.18	0.19	0.20

For determinations of absolute pressures considerably above atmospheric, the barometer reading as observed is usually sufficiently accurate. But when low absolute pressures are to be determined, such as the exhaust pressure of a condensing steam turbine, all barometer corrections should be applied. Barometer corrections for temperature, gravity, and elevation are given in Tables II, III, and IV. The A.S.M.E. Power Test Codes

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[1] For a complete discussion on barometers see *Bur. Standards, Circ.* 46; or A.S.M.E. Code, Part 2.

TABLE III.—BAROMETER CORRECTIONS FOR GRAVITY

To correct the observed reading of a mercury barometer or U-tube to the equivalent reading at standard gravity subtract (or add) the following values in inches of mercury

North latitude, deg	Elevation, ft							
	0	0	2,000	2,000	4,000	4,000	6,000	6,000
	Height of column, in. Hg							
	30	28	28	26	26	24	24	22
25	−0.05	−0.05	−0.05	−0.05	−0.05	−0.05	−0.06	−0.05
30	−0.04	−0.04	−0.04	−0.04	−0.05	−0.04	−0.05	−0.04
35	−0.03	−0.03	−0.03	−0.03	−0.03	−0.03	−0.04	−0.03
40	−0.02	−0.01	−0.02	−0.02	−0.02	−0.02	−0.03	−0.02
45	−0.00	−0.00	−0.01	−0.01	−0.01	−0.01	−0.01	−0.01
50	+0.01	+0.01	+0.01	+0.01	−0.00	−0.00	−0.00	−0.00

TABLE IV.—BAROMETER CORRECTIONS FOR ELEVATION

To correct the observed reading of a mercury barometer or U-tube to the equivalent reading at a higher elevation, subtract the following values in inches of mercury for each 100-ft difference in elevation (add for lower elevation)

Mean elevation, ft	Mean atmospheric temperature, °F						
	−20	0	20	40	60	80	100
0	0.13	0.12	0.12	0.11	0.11	0.10	0.10
1,000	0.12	0.12	0.11	0.11	0.10	0.10	0.10
2,000	0.12	0.11	0.11	0.10	0.10	0.10	0.09
3,000	0.11	0.11	0.10	0.10	0.10	0.09	0.09
4,000	0.11	0.10	0.10	0.10	0.09	0.08	0.08
5,000	0.10	0.10	0.10	0.09	0.09	0.08	0.08
6,000	0.10	0.10	0.09	0.09	0.08	0.08	0.08
7,000	0.10	0.09	0.09	0.09	0.08	0.08	0.08

require these three corrections for accurate work, and require in addition a barometer calibration correction. The Code specifies that this **barometer calibration** is to be made by correcting the observed barometer reading (Tables II and III) and reducing the result to sea level (Table IV). The difference between this result and the "weather map reading" for the location of the observed barometer, obtained from the U. S. Weather Bureau, is the calibration correction. **Barometric pressures reported by the Weather Bureau** have already been corrected to 32° and reduced to sea level. The reading of a l arometer for purposes of calibration should of course be ma the same time that the Weather Bureau takes its readings. e readings are taken at 8 A.M. and 8 P.M., 75th meridian tin S.T.).

Following is a typical example of a barometer correction:

Required: Corrected barometer reading at the center line of the turbine casing in a power plant in Boston, Mass.

Observations: Actual barometer reading 29.80 in. Hg. Barometer temperature 75°F. Barometer cistern located 16 ft below turbine center line.

Corrections:

Temperature correction, from Table II........	−0.13 in. Hg
Gravity correction, from Table III...........	−0.01 in. Hg
Elevation correction, from Table IV..........	−0.02 in. Hg
Calibration correction determined from Weather Bureau comparison as already described (assumed in this case).............	+0.04 in. Hg
Total net correction.......................	−0.12 in. Hg

Results: The barometric pressure at the turbine center line is 29.68 in. Hg or 29.68 × 0.491 = 14.58 lb per sq in. abs (see Table I).

Aneroid Barometer.—The aneroid (without moisture) employs a sensitive diaphragm or flexible chamber which communicates differences of pressure through a multiplying mechanism to an indicating needle. The dial is graduated to correspond with the readings of a mercurial barometer. This type, being both convenient in size and portable, has come into very general use. The aneroid can be made more sensitive (*i.e.*, show smaller changes of atmospheric pressure) than the mercurial barometer. Unfortunately, however, it does not preserve its accuracy for a great length of time; hence it must be adjusted from time to time to a standard mercurial barometer.

The aneroid barometer is also made in the more accurate null type in which the diaphragm is brought back to an initial position by changing the tension of the loading spring. This same motion rotates the scale to give the new reading.

Bourdon Pressure Gages.—For the higher pressures, Bourdon gages are employed in the place of manometers. These gages are ordinarily made for pressures up to 2,500 lb per sq in., though special gages may read to 10,000 lb or even higher.

In the Bourdon gage (Fig. 12), pressure variations are indicated and measured through a multiplying mechanism attached to the extreme end of a circular flattened tube of elliptical cross section. This elliptical tube when bent in the form of a circle and subjected to an internal pressure will tend to straighten out, owing to the tendency of the elliptical section to become circular. This movement will, in turn, actuate the multiplying device in amount proportional to the pressure change. The multiplying mechanism consists of one or more levers actuating a segmental rack as shown, which in turn meshes with the pinion of the pointer shaft.

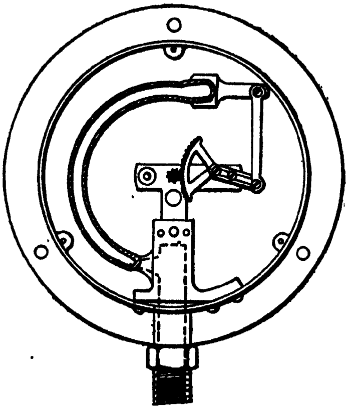

FIG. 12.—Mechanism of Bourdon gage.

Diaphragm and bellows gages are used mainly for low pressures, though metallic-bellows gages are available for pressures as high as 3,000 lb per sq in. Compared with the bent-tube gage, a diaphragm or bellows gage provides a larger force for actuating the indicating or recording mechanism, owing to the larger area upon which the pressure acts. Hence these gages are especially suitable for measuring pressures in the manometer range or in the low Bourdon type ranges. **Slack-diaphragm gages** use soft elastic diaphragms of leather, treated cloth, or rubber, externally spring loaded. Several commercial makes of draft gages operate on this principle, and they have the advantage of not involving any liquid or requiring accurate leveling. Like any spring gage, however, they are subject to calibration errors.

The Bourdon or diaphragm pressure gage may be made self-recording by means of clockwork attachments which rotate a graduated dial chart, upon which the pressure indicator bears and continuously records the pressure. To determine the average pressure from one of these charts an integrating instrument is used. The usual recording pressure gage has the pencil recorder rigidly attached to the end of the Bourdon tube. This tube is bent in helical form so as to impart sufficient free motion to the pencil recorder and thus eliminate the multiplying links.

Electric gages are available for both pressure and vacuum, in which the movement of a diaphragm or piston is magnified by the use of a bridge or an electronic device. Such gages may be either indicating or recording, and they may be used for rapidly varying pressures by the application of an oscillograph.

EXPERIMENT 1. Measurement of Static and Velocity Pressures by Pitot Tube and Manometers

I. PREFACE

The primary objects of this experiment are to illustrate the distinction between static pressure and velocity pressure and to show how each is measured. Incidentally, the experiment serves two other useful purposes, *viz.*, it offers an opportunity for the study and use of the various types of manometers previously described, and it gives the student an idea of the performance of a centrifugal fan.

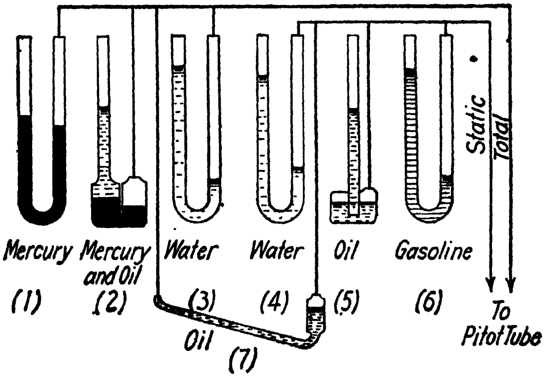

FIG. 13.—Manometer connections for measurement of fan pressures in Experiment 1.

II. APPARATUS

In order to introduce a variety of instruments, the gages to be used in this experiment are to conform to the descriptions given below and are to be connected as shown by Fig. 13. Seven gages are used in this case, three indicating total pressure, three static pressure, and one velocity pressure. For convenience these gages have been numbered and the following description applies to each gage as numbered.

1. Reads total pressure in inches of mercury. Simple U-tube manometer filled with mercury.
2. Reads total pressure in ounces per square inch. Two-fluid U-tube manometer using mercury in one leg and oil of a known specific gravity above the mercury in the other (Bacharach).
3. Reads total pressure in inches of water. Simple U-tube manometer filled with water.
4. Reads static pressure in inches of water. Simple U-tube manometer filled with water.
5. Reads static pressure in inches of water. Cistern or well type manometer filled with oil but graduated in inches of water. Change of zero level in the cistern is compensated for in the calibration.
6. Reads static pressure in inches of gasoline. Simple U-tube manometer filled with gasoline.
7. Reads velocity pressure in inches of water. Inclined manometer or "draft gage" filled with oil but calibrated in inches of water.

The pressures to be measured are produced by a small high-pressure centrifugal fan. The fan is fitted with test pipe on the discharge, and provision is made for mounting throttling orifices of various sizes at the end of the test pipe.

The following apparatus should be secured from the instrument room: (*a*) pitot tube with rubber tubing; (*b*) centrifugal tachometer; (*c*) data board; (*d*) 120-deg (or 220-deg) mercury thermometer.

III. INSTRUCTIONS

1. Procedure.—Record the zero reading of each gage. (Set scale to zero if adjustment is provided.)

Connect the pitot tube and insert same in the discharge pipe, pointing upstream. Place tube in the exact center of the pipe and clamp firmly in position.

Take readings on all gages with discharge pipe full open; observe also the fan speed and the air temperature. These values should be kept constant throughout the test, if possible. Repeat observations with each size of orifice and with no discharge (blanked pipe). As an alternate test, the fan speed may be varied and the orifice area kept constant. Allow ample time in taking readings, to secure a good average if gage readings are fluctuating. Measure accurately the internal diameter of test pipe and the diameter of each orifice.

Check zero readings of all gages after test is completed.

Form 1.—Measurement of Fan Pressures—Log Sheet

Barometer.........Diameter of test pipe.. .. ..Area of test pipe.......

Run number........................	1	2	3	4	5	6
1. Diameter of orifice, in...............						
2. Percentage of full outlet.............						
3. Fan speed, r.p.m.....................						
4. Air temperature, deg. fahr............						
Manometer 1, total pressure, in. mercury .						
5. Left leg, in.........................						
6. Right leg, in........................						
7. Difference, in.......................						
Manometer 2, total pressure, oz. per sq. in.						
8. Zero reading, oz.....................						
9. Total reading, oz....................						
10. Difference, oz......................						
Manometer 3, total pressure, in. water....						
11. Left leg, in.........................						
12. Right leg, in........................						
13. Difference, in.......................						
Manometer 4, static pressure, in. water....						
14. Left leg, in.........................						
15. Right leg, in........................						
16. Difference, in.......................						
Manometer 5, static pressure, in. water....						
17. Zero reading, in.....................						
18. Total reading, in....................						
19. Difference, in.......................						
Manometer 6, static pressure, in. gasoline						
20. Left leg, in.........................						
21. Right leg, in........................						
22. Difference, in.......................						
Manometer 7, velocity pressure, in. water..						
23. Zero reading, in.....................						
24. Total reading, in....................						
25. Difference, in.......................						

Date............... Observers:.................................

....

....

2. Report.—Using the readings of gages 3, 4, and 7, prepare a tabulation showing for each size of orifice the comparison of velocity pressure as read (gage 7), with the difference between the total pressure and the static pressure.

Using the readings of the same gages, plot the following curves:

1. Total pressure against orifice area. The latter is to be expressed in percentage of pipe area (*not* in square inches or square feet) and will be plotted as abscissas. This quantity, *i.e.*, the orifice area in percentage of pipe area, is often called the "percentage of full outlet."
2. Static pressure against percentage of full outlet.
3. Velocity pressure against percentage of full outlet.

If the alternate test is run, using various fan speeds and a constant orifice size, the same three curves will be plotted, except that rpm instead of orifice area will be the abscissas. In this case logarithmic coordinates should be used, and all curves will become straight lines, with a slope of 2 (see Fan Laws, Table XXVI, page 324).

Calculate from the observed readings the average conversion factor by which the readings of gages 1, 2, and 6 are to be multiplied to convert them to inches of water. Compare these average values with the corresponding conversion factors calculated from the densities of the liquids used and from the units involved.

EXPERIMENT 2. Calibration of Bourdon Pressure Gage

I. PREFACE

Read section on Bourdon Pressure Gages, page 22.

II. APPARATUS

Procure a pressure gage and a dead-weight gage tester from the instrument room.

Take off the rim from the gage and remove glass; use jack to remove indicating hand, then remove face. Note the relative positions and details of the internal mechanism and make diagrammatic sketches of same. Discuss method of securing multiplication and adjustment of the multiplying mechanism. Low-pressure gages should be calibrated by comparison with a standard mercury column. Attach both to the same pressure drum and read the gages simultaneously (see Experiment 3, page 31).

For calibrations not to exceed 300 lb maximum pressure per square inch, the weighted plunger or **dead-weight tester** similar to Fig. 14 is universally employed. The piston of the plunger is exactly 0.2 sq in. in area. The plunger and table weigh 1 lb. The accurate weights provided are made up of one 1-lb, one 2-lb, and sufficient 4-lb weights to give the full capacity of 300 lb. Dead-weight testers, with or without lever multiplying systems, are also available for testing very low pressure gages (below 5 lb) and very high pressure gages (up to 15,000 lb).

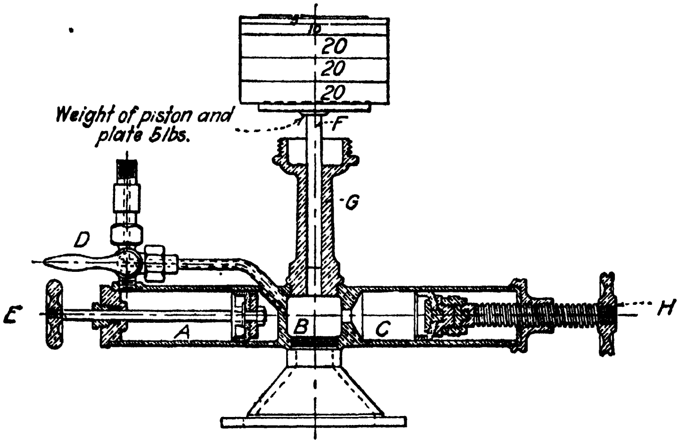

Fig. 14.—Dead-weight gage tester (American).

The type of dead-weight tester shown in Fig. 14 is equipped with both a pressure chamber *C* and an oil-storage chamber *A*. Many testers do not have an oil-storage chamber, thus making it necessary to drain the oil before replacing the instrument in its case. The following instructions for the use of the apparatus refer to the type illustrated in Fig. 14, and, after these instructions have been studied, the procedure for using the simpler apparatus without an oil-storage reservoir will be obvious.

To use the apparatus, see that chambers *A*, *B*, and *C* are filled with a light mineral oil. To fill chambers *B* and *C* (assuming cock handle *D* is in position as shown), remove plunger *F*, then screw in piston *C* by means of hand wheel *H* until it strikes the lower stop or to a point where oil will rise into the cup at top of plunger cylinder *G*. As oil is being introduced through cylinder *G*, screw out on piston *H*, maintaining oil level in cup until piston *H* strikes the outer stop or until the desired amount of oil is drawn into cylinder *C*. The oil is kept at the cup level so as to avoid trapping air into the apparatus.

Owing to its elasticity, sufficient air may be trapped in the chambers to permit the piston in cylinder *C* to reach its inner stop before the full capacity of the gage being tested is reached.

To fill chamber *A*, pull out on handle *E*, turn handle of cock *D* to right; this opens communication through tube leading from chamber *B* through cock *D* and into cylinder *A*. As oil is supplied through cylinder *G*, push in handle *E*, maintaining oil level in cup, until the inner piston stop is reached or the desired amount of oil is drawn into cylinder *A*.

After observing zero reading of gage, attach gage to instrument by special connection above cock *D*. Insert plunger, screw out on handle *H* until table of plunger rests at about 1 in. above top of cylinder *G*. This level should be maintained by screwing in on handle *H* as the calibration pressures increase. Turn handle of cock down, pull out on handle *E* until gage shows pressure and gage tube is sealed with oil, then turn handle of cock *D* to the left which connects gage with chamber *B*.

Rotate plunger and table *F* by hand (to destroy the vertical component of cylinder friction). The gage, if correct and free from backlash, should read 5 lb pressure and for each addition of 1 lb avoirdupois placed on the plunger table the gage should show an increased pressure of 5 lb in its reading.

III. INSTRUCTIONS

1. Procedure.—Check the gage at each 5-lb increment, taking two readings at each pressure, as follows: (1) spin table and supported weights gently and apply **downward** pressure (with pencil or otherwise) at the axis of rotation, releasing this pressure **slowly**, and read gage while the weights are still moving. Tap gage smartly with pencil or finger before reading, and estimate reading to one-tenth scale division; (2) repeat the process, this time applying an **upward** pressure underneath the weight platform. On account of the backlash in the gearing and looseness of pivoted joints of the gage, the rising and falling calibrations may not be exactly the same. In practice, the mean value is taken as the correct gage reading.

If the mean of the rising and falling calibrations differs by more than 1 lb from the standard or true reading, the gage hand should be removed and reset. After the gage hand is reset, the calibration should be repeated. After calibration and before removing gage, turn handle of cock *D* down, push in slightly on handle *E*, *i.e.*, sufficiently to withdraw oil from gage tube and connections into chamber *A*. The gage may then be removed.

Plot curves between error of gage in pounds or percentage as ordinates, and pressure in pounds per square inch as read from the gage as abscissas, plus errors being plotted above the axis of abscissas and minus errors plotted below the axis.

Form 2.—Calibration of Pressure Gage by Dead-weight Tester

Type of gage............................Laboratory No..........
Maker...
Range.........................Graduation increment..............

Pressures in pounds per square inch				Corrections	
Actual, by dead weights	Gage readings			In gage units	Per cent
	Increasing	Decreasing	Mean		

Date................ Observers:..............................
....
....

Percentage error of gage

$$= \frac{\text{error of gage reading in pounds per square inch}}{\text{gage reading in pounds per square inch}}.$$

IV. PRECAUTIONS AND NOTES ON PRESSURE MEASUREMENTS

1. To protect the spring and the multiplying mechanism of a Bourdon gage against high temperatures, a U- or pigtail tube (Fig. 15) should be used when the gage is connected to a steam line. This tube retains the condensation and provides a water seal between the gage tube and the source of live steam. Care should be taken to note that all joints above the water

seal are tight; otherwise the water of the seal will leak out and live steam be admitted to the gage tube and overheat the mechanism.

2. Gages should be protected from undue vibrations which produce wear of the mechanism and destroy the sensitive adjustments. When water hammer or machine pulsations produce a jerky motion or vibration of the indicating pointer, the gage cock should be throttled slightly, so as to dampen the pulsating effect on the indicating hand and thus secure an average reading of the pressure fluctuations. Care must be taken not to throttle the cock too much. Gage protectors or vibration dampeners of special design may be procured from certain of the instrument companies.

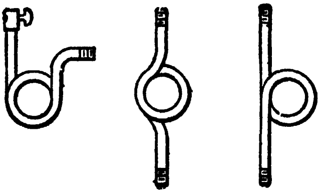

FIG. 15.—Water-seal tubes for steam gages.

3. Where vibrations are unavoidable, use double-tube gages.

4. Gage and manometer connections for static pressure measurement should be made flush with the inside of the chamber or pipe and should be free from burrs or irregularities (see page 273).

5. The gage selected for a given purpose should have a range of approximately one and one-half times the normal working pressure.

In standard tests, the gages are calibrated both before and after the test. This is especially necessary when the gages are subjected to variable or pulsating pressures.

6. Before attempting to reset the hand on a gage, determine whether or not the gage has already been used on test. Reset the hand only in case the test has not yet been started, or after a full calibration has first been secured.

7. Gages are often calibrated or checked against standard gages by attaching both to the same pressure drum and reading the gages simultaneously. Form 3 is to be used for such a calibration. The columns headed "Actual" will have identical values. From the correction curves of the standard gage the true pressure reading may be obtained by changing the sign of the error and adding the correction for the given pressure to the standard-gage reading. The error of the gage being tested may then be found and its calibration curves plotted.

EXPERIMENT 3. Calibration of Bourdon Vacuum Gage

I. PREPARATION AND INSTRUCTIONS

Since the vacuum-gage dial is graduated to read in inches of mercury, it is most simply calibrated by direct comparison with a mercury column. One of a number of simple devices that may be used is illustrated in Fig. 16. It consists of a small tank or reservoir to which a standard mercury column G and gage C to be tested are attached. The vacuum is secured in this case by exhausting the air from the tank by means of the simple water ejector *A*. Regulation so as to obtain different degrees of vacuum may best be secured by having the water-supply valve full open and regulating vacuum by admitting air to reservoir through the needle valve *F*.

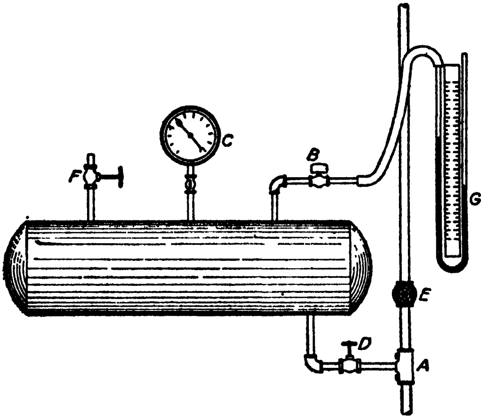

Fig. 16.—Vacuum-gage calibration apparatus.

Having selected the pressure-reduction increment for the calibration, a full range set of readings for both a rising and a decreasing vacuum should comprise the data for a complete calibration.

Tap gage case lightly during calibration to ensure that the indicating mechanism is functioning freely.

Before any data are taken, the system should be checked for tightness over a time period by observing any change in manometer reading under high vacuum, valve *D* closed.

If the gage is more than 1 in. in error, the hand should be reset and the gage recalibrated. Consult notes under calibration of pressure gage (Experiment 2, page 29).

Results.—For tabulation, use Form 3.

Plot curves between error of gage in inches of mercury or per cent as ordinates and gage reading as abscissas, plus errors to be plotted above axis of abscissas and minus errors plotted below axis.

$$\text{Percentage error} = \frac{\text{error in gage reading, inches of mercury}}{\text{actual gage reading, inches of mercury}}.$$

Form 3.—Calibration of Pressure or Vacuum Gage by Comparison with Standard

General description

Type.................................... Laboratory No.........

Maker...

Range........................ Graduation increment...............

Room temperature..........deg. fahr. Barometer.............in. Hg.

Standard gage					Tested gage				
Pressures					Pressures				
Gage			Correction	Actual	Gage			Actual	Correction
Up	Down	Mean			Up	Down	Mean		

Date............... Observers:...................................

....

....

II. NOTES

In many cases the ejector is displaced by a hand or power-driven suction pump.

In order to guard against drawing water back into the drum and sometimes over into the leg of the mercury manometer, valve *D* should be shut before any sudden shutoff of ejector supply water. Air may then be admitted into the tank through valve *F* and the vacuum destroyed. A horizontal check valve at *B* is sometimes used.

TEMPERATURE MEASUREMENTS

Experiment 4

I. PREFACE

The engineer's need for accurate temperature measurement under a large variety of conditions has been met by the development of the several types of instruments presented in Table II.

The human sense of touch was of course the earliest and still is the most commonly used thermometer. Similarly the eye, the earliest pyrometer, is still relied upon to a surprising extent in industrial work, particularly in the working of steel. In considering the accuracy of the touch thermometer one has only to remember that, when coming in with cold hands on a wintry day, a basin of tepid water feels almost hot. To the moist skin a "draft" of warm dry air feels "cold." Nor does it take much experience in the forge shop to demonstrate the effect of surrounding light conditions on the indications of the original "optical" pyrometer. Proper forging "heat" looks quite different in a dark corner from what it does in the full sunlight near the window. It is obvious that in almost all engineering work far greater accuracy than this is required.

Measurement of temperature is generally supposed to be one of the simplest and most accurate measurements performed in engineering. This is decidedly a misconception. Accurate temperature measurement under certain conditions is *impossible* with our present knowledge. Errors in temperature measurement are frequently not due to the instruments themselves but rather to the way in which the instruments are installed and used. For this reason, the present article deals not only with the calibration of thermometers and pyrometers but also with the right and wrong methods of using these instruments.

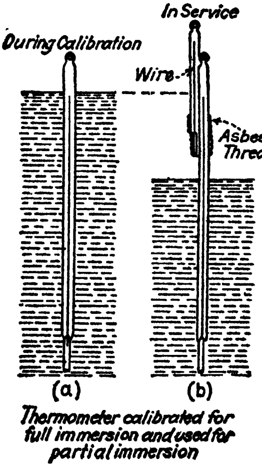
In Service
During Calibration
Wire
Asbestos Thread
(a)
(b)
Thermometer calibrated for full immersion and used for partial immersion

FIG. 17.—Method of correcting full-immersion thermometer for stem emergence.

TABLE V.—CLASSIFICATION AND LIMITATIONS OF TEMPERATURE-MEASURING INSTRUMENTS

Common name	Type or subclass	Principle of operation	Usual ranges of temperature measured, deg. fahr.	Special advantages	Typical engineering and industrial uses	Probable accuracy (deg. fahr. plus or minus)
Mercury-thermometer	Common	Volumetric expansion of a liquid in a vacuum	−35 to 600	Inexpensive; easy to handle; easy to read; accurate	Water temperatures in power-plant test work	Depends on design
Mercury thermometer	Gas-filled	Volumetric expansion of a liquid under pressure	0 to 1000	Wide range; simple; inexpensive; accurate	Steam temperatures; flash point of oils; flue-gas temperatures	Depends on design
Expansion thermometer	Liquid, gas or vapor filled, pressure-gage type	Pressure change of liquid, gas or vapor confined in constant volume	50 to 1000	Rugged; recorder or indicator may be placed at a distance (to 200 ft.)	Flue-gas, steam, and water temperatures in power plants; core-oven, baking-oven, and enameling-oven temperatures	2 to 10
Electrical-resistance thermometer	....	Change of electrical resistance of a metal with change in temperature	−300 to 1800	Accurate; sensitive; one indicator for many temperatures; distant reading	Temperatures within stored materials subject to heating, as coal, grain, etc.; laboratory and experimental work	0.1 to 0.5

Thermocouple pyrometer	**Base-metal couple and millivoltmeter indicator**	Measurement of thermocouple e.m.f. by millivoltmeter	To 200 with Bi-Sb; To 900 with Cu-constantan; To 1500 with iron-constantan;	Rugged; well adapted to distant reading; easily made recording; one indicator for many couples	Temperatures in furnaces of all kinds, up to 2200 deg. fahr.; heat-treating temperatures; temperatures in glass, ceramic, and oil-refining industries; temperatures in boiler settings and flues; exhaust temperatures of gas and oil engines	3 to 20
Thermocouple pyrometer	**Base-metal couple and potentiometer indicator**	Thermocouple e.m.f. balanced against known battery e.m.f.	To 1600 with nichrome-constantan; To 2200 with chromel-alumel; To 2200 with nichrome-alumel	More accurate than millivoltmeter type		0.5 to 10
Thermocouple pyrometer	Noble-metal couple and potentiometer indicator	Thermocouple e.m.f. balanced against known battery e.m.f.	To 2600 with platinum-rhodium	High range; temperature-e m.f. curve does not vary with age of couple	High-temperature or precision work in laboratory	0.5 to 5
Radiation pyrometer	....	Total radiation from hot body heats thermocouple placed in focus of concave mirror	From red heat to maximum temperatures encountered in industrial work	High and wide temperature range; can be made recording	Temperatures of furnace walls and interiors, large ingots; molten metals, glass	10 to 50 under black-body conditions
Optical pyrometer	....	Sighting tube with disappearing filament or divided field compares intensity of monochromatic radiation from hot body with that of standard	From red heat to maximum temperatures encountered in industrial work	Easy to use; high and wide temperature range; instrument may be "sighted" on small hot body or body in motion	Temperatures of furnaces, ingots, forgings, steel plates and rails, ceramic kilns, molten metals	10 to 35 under black-body conditions
Fusion pyrometer	**Pyrometric cones**	Series of fusible cones with melting points about 50 deg. apart	1000 to 3600	Simple; inexpensive; wide range	Ceramic kilns	20 to 50

1. Liquid-in-glass thermometers are usually filled with mercury and have a vacuum in the capillary. Since the freezing point of mercury is −38°F, alcohol or pentane are used for very low temperatures. To measure temperatures above 600°F, the mercury is sealed under pressure, using nitrogen or carbon dioxide in the capillary. With special glass this gives a range up to 1000°F, although difficulties from stem distortion may be encountered above 900°F. For greater ease in reading, the glass stem may be made with colored inserts, colored background, or with a magnifying-lens front. *Glass-stem thermometers are graduated for complete immersion* of bulb and stem unless partial immersion is specified. When the stem of a total-immersion thermometer is only partly immersed, as in a thermometer well, a **correction for stem exposure** must be made. The A.S.M.E. Test Codes recommend the attachment of a second thermometer for securing stem temperature, as shown in Fig. 17, and corrections made by the formula:

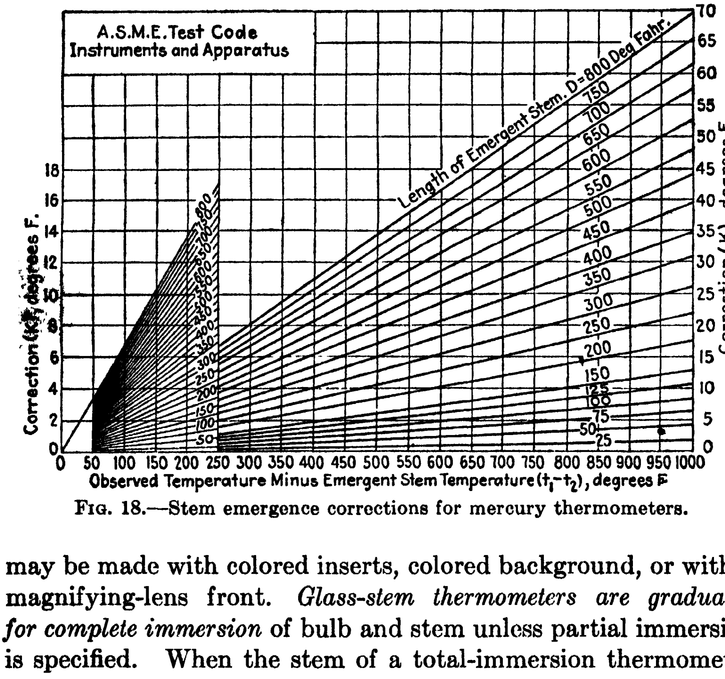

FIG. 18.—Stem emergence corrections for mercury thermometers.

$$K = 0.000088D(t_1 - t_2), \tag{2}$$

where D = number of degrees of exposed filament.

t_1 = reading of main thermometer.

t_2 = reading of attached thermometer.

When the stem is cooler than the bulb, the correction K, in degrees F is added. When the stem is warmer than the bulb, the correction is subtracted. Figure 18 is a handy correction chart based on Eq. (2).

Glass thermometers may be purchased in armored or protecting cases for either angle or vertical mounting (Fig. 19) and in over-all lengths of 3 ft or more. In certain of these types, the glass bulb is sealed from the fluid by the metal case; in others, the protecting covering is simply a screen or perforated metal sleeve loosely fitted around the thermometer bulb. The latter type permits the fluid surrounding the thermometer to come into actual contact with the glass bulb. The protecting case of the sealed type introduces a temperature lag which may be objectionable for a certain class of work. The use of thermometer wells (Fig. 20) likewise introduces a lag effect.

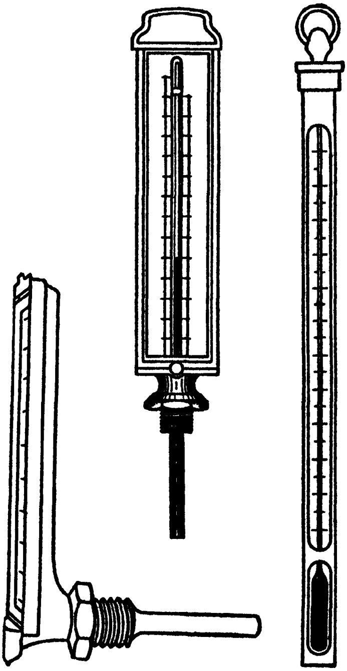

Fig. 19.—Commercial armored thermometers.

Thermometer wells similar to the finned well of Fig. 20 are prescribed by the A.S.M.E. Codes for measuring the temperature of a gas or of a superheated vapor in a pipe. When the pipe contains a saturated vapor only, or a flowing liquid, the fins may be omitted. Thermometer wells should be filled with a nonviscous liquid of high conductivity. Water, alcohol, or kerosene may be used for low temperatures, mercury or oil for higher temperatures, and tin or solder for temperatures above 600°F.

Flue-gas temperatures of a commercial steam boiler may be checked by the use of the high reading mercury thermometer. To introduce the thermometer into the center of the stream of flue gases (this distance may be 6 ft or more from the side wall of boiler setting), the following method may be used (see Fig. 21). In preparation, secure a piece of ¾-in. standard black pipe of sufficient length to reach the center of the gas stream and extend several feet beyond the outside wall of the setting. A milling cutter should be used to remove the outside face of the pipe to a depth close to the center of the pipe. The cut should be somewhat longer than the thermometer to be used and its location along the pipe such as to permit its placement into the center of the gas stream. In the milled opening, a piece of shredded asbestos wicking is placed to form a cushion upon which the thermometer rests. After inserting this device in the boiler setting, sufficient time must be allowed for the rod and thermometer to reach the temperature of the surrounding gases. If the pipe is then drawn out quickly and the thermometer read, a very close determination of true gas temperature at the center of the gas stream may be secured. Very little drop in temperature will result from traversing cooler sections of the gas stream if the withdrawal is done expeditiously. The iron rod, asbestos pad, and thermometer will lose very little heat during the short period required for the withdrawal and reading of the temperature.

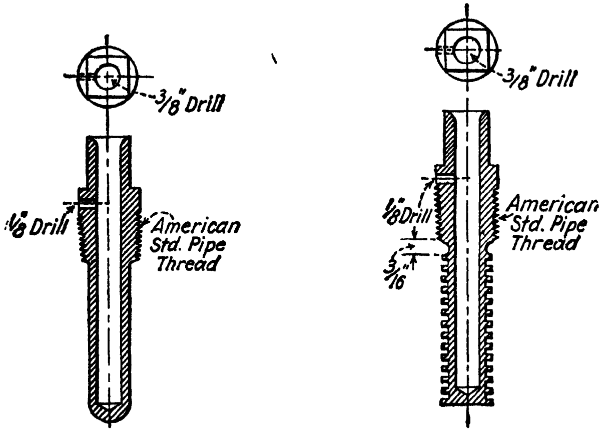

FIG. 20.—Thermometer wells. (*A.S.M.E. Code.*)

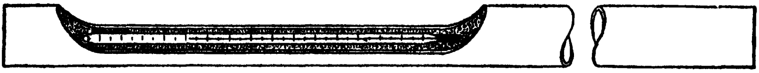

FIG. 21.—Taking flue-gas temperatures with a common thermometer.

2. Solid-expansion thermometers are of importance mainly because of their wide use in thermostat controllers, although laboratory and industrial thermometers of this type are also available. The sensitive element is usually a bimetallic strip or coil, fixed at one end and carrying indicator or contacts at the other. For opening and closing electrical contacts a positive snap-action device is included. The quality of these solid-expansion thermometers covers a wide range, from the domestic oven, refrigerator, and room thermometers costing but a few cents, to laboratory and industrial thermometers of fair precision.

3. Fluid-expansion thermometers are widely used in industrial service because they are the least expensive of the distant-reading and the recording types. The operating fluid may be a gas, a vapor, or a liquid, and the fluid bulb is connected by capillary tubing to a pressure-spring displacement unit, as shown in the line sketch (Fig. 22). The range and accuracy of a fluid-expansion thermometer depend upon its design and construction. Inexpensive units are used for automobile instrument boards, with an accuracy of perhaps 5 to 10 per cent of full-scale reading; whereas larger units for industrial and laboratory service may have a maximum error of 1 to 3 per cent of full-scale reading. When the engineer uses the fluid-expansion thermometer, he should keep in mind four conditions that may cause errors in its indications: (1) If the capillary and the spring tube of a gas- or liquid-filled unit are subjected to ambient temperatures widely different from that for which the instrument was calibrated, a larger error may result. The vapor-pressure type (Fig. 22) is not affected by these variations. Compensating devices are also available to correct this condition in the gas and liquid types. (2) Since the sensitive bulb of a fluid-expansion thermometer is comparatively large in size, the instrument is subject to radiation effects when the bulb is used in air or gas. These radiation errors are minimized through the use of stainless-steel bulbs or sockets, but the best remedy is to locate the bulb where it will not "see" surfaces much hotter or colder than the air or gas temperature being measured. (3) If temperatures beyond the range of the instrument are likely to be encountered, some type of over-range protection should be built into the instrument. (4) In recording instruments, the drag of the pen or stylus may affect the accuracy and tend to make the record lag behind the actual temperature variations.

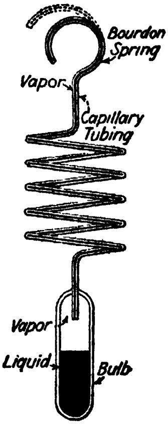

Fig. 22.—Elements of a vapor-pressure thermometer. (*A.S.M.E. Code.*)

In the liquid type of fluid-expansion thermometer, mercury may be used for range −38 to 1000°F and alcohol −50 to 300°F. The gas-filled type employs nitrogen with a range −60 to 1000°F. For the vapor-filled type, various liquids are used with an overall range of −20 to 700°F.

4. Electrical-resistance Thermometers.—These instruments depend for their operation upon the change in electrical resistance produced in a given metal when the metal is subjected to a *change in temperature.*

The instrument incorporates in its construction a coil of nickel, platinum, or copper wire (instead of a thermocouple, to be described later). As the temperature of the coil rises, its electrical resistance increases. If, then, the law of change of these two variables is known, the temperature may be determined for a given measured resistance. This resistance may be measured by any one of three methods, Wheatstone bridge, potentiometer, or deflection galvanometer.

The electrical-resistance thermometer is very sensitive and highly accurate. It is used as the international standard in the range from −190 to 660°C. A single instrument may serve as an indicator for a series of coils located at distant points through a proper switch arrangement.

The resistance thermometer has the further advantage of being adjustable in scale and range by the adjustment of the resistances in the bridge. The scale need not include zero; and it may be spread out, so that a range of perhaps 25° covers the entire length of the scale, say 6 in. Thus the absolute accuracy may be increased to a small fraction of 1 per cent, depending upon the calibration and upon the sensitiveness of the galvanometer. One disadvantage of the resistance thermometer is its rather high cost, and its applications are usually limited to distant-reading installations in which precise measurements are needed or where the range covered is only a few degrees, as for instance in the accurate control of temperatures in rooms or storage spaces.

5. Thermocouples are pairs of wires, of dissimilar metals, connected at both ends. When the two junctions are subjected to different temperatures, an electrical potential is set up between them. This voltage is almost directly proportional to the temperature difference, and hence a voltage-measuring instrument placed in the circuit will measure temperature. Table VI gives the emf generated at various temperatures with commonly used thermocouples. The couples are not recommended for use above the maximum temperatures indicated. Any convenient size of wire may be used, but a good electrical contact at the junctions and insulation of the rest of the wire are important. Junctions may be merely twisted together, but, for mechanical strength and to ensure good electrical contact in spite of oxidation, welded or soldered junctions are preferable. Insulations may be enamel, silk, cotton, or other fiber for low temperatures, and asbestos, porcelain, or other refractory for high temperatures. Pure metals *should be used for thermocouple wires, and each batch of wire should be calibrated. Calibrated wire of various sizes and with* various kinds of insulation may be obtained from the instrument manufacturers or supply houses.

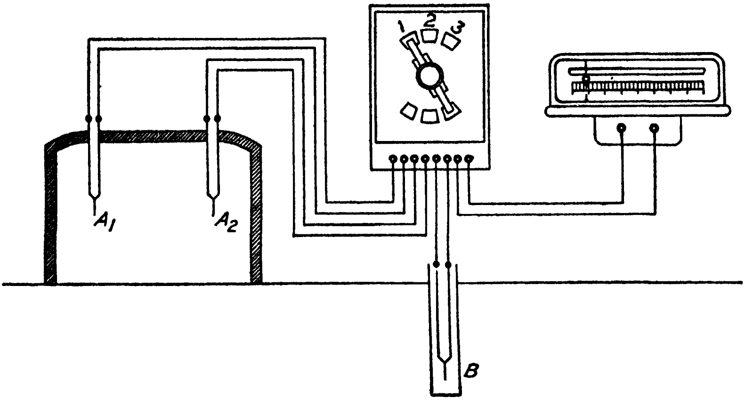

FIG. 23.—Elements of a commercial thermocouple installation for measurement of furnace temperatures.

In order to complete the electrical circuit through an instrument (Fig. 23) two or more junctions of dissimilar metals are always necessary, and the true temperature of the "hot junction" *A* cannot be known unless the temperature of the "cold junction" *B* is held at a constant and known temperature. Various means are used for maintaining the cold-end thermocouple junction at a constant fixed temperature. For accurate work a thermos bottle containing cracked ice is used; or the cold-end junction may be buried in the ground to a depth of 10 ft or more, where local temperature changes are not present (*B*, Fig. 23). Internal compensating devices built into the instrument, that automatically adjust the cold end for varying conditions, now have the preference over other methods in commercial testing.

Many combinations of metals have been tested to determine their value as thermocouple elements. The ranges for each of the common couples are given in Table V, page 34, and typical millivolt-temperature calibrations are given in Table VI. Other advantages of the various combinations are:

TABLE VI.—TEMPERATURE-MILLIVOLT RELATIONS FOR THERMOCOUPLES

°F above cold-junction temp.	Millivolts			
	Copper constantan	Iron constantan	Chromel alumel	Platinum 10% rhodium
25	0.55	0.72	0.53	0.071
50	1.11	1.44	1.08	0.148
75	1.67	2.16	1.63	0.229
100	2.23	2.88	2.20	0.313
150	3.40	4.36	3.34	0.493
200	4.65	5.84	4.50	0.687
250	5.93	7.36	5.65	0.892
300	7.23	8.87	6.77	1.108
350	8.60	10.40	7.88	1.333
400	9.99	11.94	8.99	1.565
450	11.40	13.47	10.11	1.804
500	12.87	15.01	11.24	2.048
600	16.03	18.08	13.53	2.549
700		21.15	15.86	3.067
800		24.23	18.20	3.597
900		27.33	20.56	4.136
1000		30.46	22.93	4.686
1200		36.96	27.66	5.817
1400		43.88	32.33	6.990
1600		50.92	36.88	8.204
1800		57.96	41.30	9.457
2000		65.00	45.57	10.749
2500			55.53	14.060
3000				17.339

Iron constantan is the most commonly used combination for moderate temperatures, owing to its cheapness. (Constantan is 60 per cent copper, 40 per cent nickel.) This couple generates a higher emf than any of the other common units, and hence a less sensitive instrument is required. The couple will retain its calibration even though heavily oxidized.

Copper constantan also generates a high potential, and the purity of the commercial wires is high. It is preferred over iron constantan for low-temperature work, because the wiring is simplified owing to the common use of copper in all electrical circuits. The entire circuit is copper with the exception of the constantan wire leading from the hot junction to the cold junction.

Chromel alumel resists corrosion and maintains its calibration at high temperatures; it is generally preferred over other couples for the range from 1500 to 2200°F.

Platinum-rhodium or "noble-metal" couples are the basic standard and are more reliable than any of the others on account of freedom from corrosion and constancy of calibration, but they are very expensive, and hence they can be used in small sizes only. They also require more sensitive instruments because they generate a smaller emf.

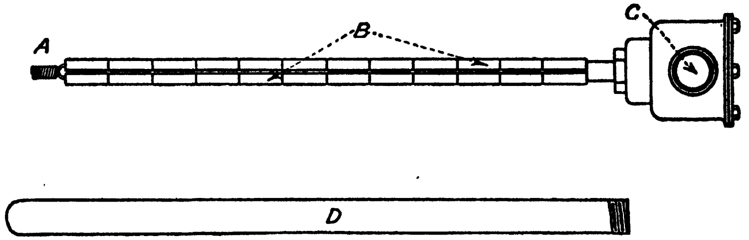

FIG. 24.—Coarse-wire thermocouple with porcelain-bead insulators, terminal head, and protecting pipe.

Other metals used for thermocouples include nichrome and stainless steel.

A satisfactory couple may be homemade, and there is nothing technical or complex in the making. The two wires are twisted together a few turns at the end (for strength) and then welded at the extreme tip. For heavy wires (Nos. 8 to 14) an oxy-acetylene flame may be used (borax flux) but for No. 16 or lighter a direct-current carbon arc is preferable. A resistance or a lamp bank is used to adjust the arc according to the wire size. Wires up to No. 40 may be satisfactorily welded in this way.

Couple wires should be insulated with porcelain beads, braided asbestos, or some type of "spaghetti" and for high-temperature work the entire assembly encased in a protecting tube of iron pipe, nichrome, porcelain, or silicon carbide (see Fig. 24).

Series connection of several thermocouples multiplies the potential and is used as a means of increasing sensitivity when temperature differences are being measured. Four pairs of junctions will increase the sensitivity four times since the instrument reading will be divided by four. *Parallel connection* of couples gives average temperatures, as for instance in Fig. 23 where the average of A_1 and A_2 is obtained by throwing the switch to 3. The internal wiring of the switch is such that the couples are then connected in parallel.

The millivolt-temperature relation for a thermocouple should not be assumed without actual calibration. Even in the case of two couples from the same batch of wire, the calibration may vary from 5 to 25°, depending on the temperature range and upon the purity of the wires. Standardized wire may be purchased from the instrument companies, however, with a guarantee of accuracy within 1 or 2 per cent (or 0.5 per cent, if required) when used with calibration curves furnished by them. This is the best procedure, unless large quantities of wire are used and price is all-important.

6. Thermocouple Instruments.—The two common instruments for measuring the electrical potential developed by thermocouples are the millivoltmeter and the potentiometer-galvanometer. Each method has its advantages, and it is important that these be understood in order that the proper selection of instrument can be made for each application.

Millivoltmeter Method.—In this case an ordinary moving-coil D'Arsonval instrument is used; in fact any ordinary electrical millivoltmeter may be substituted. The behavior of such an instrument depends on its internal resistance. A low-resistance millivoltmeter is a more sturdy unit, but its indications are affected by the resistance and the temperature of the lead wires. High internal resistance reduces instrument current and hence reduces deflection torque, and a more delicate instrument is necessary, but the readings are only slightly affected by ordinary changes in the resistance of the lead wires. The low-resistance instrument may be partly compensated for its deficiencies by special construction, but usually a millivoltmeter with at least 200 ohms internal resistance is preferred for thermocouple work.

Potentiometer Method.—In the potentiometer method of measuring the developed emf., the resistance of the leads does not affect the readings. This desirable feature is accomplished by balancing the generated potential against a known potential of a standard electric cell, provided the known potential remains constant and the fixed resistances in the instrument proper remain constant. Since in laboratory work the potentiometer is the recognized instrument for accurately measuring the generated emf, it deserves a somewhat full description. Most potentiometer circuits are closely the same and, therefore, a description of the circuit as used by one of the prominent manufacturers will suffice for this condensed treatise.

Current from the dry cell flows through the instrument circuit (Fig. 25). Owing to the slide wire *DGE*, the battery current will set up a difference of potential between *D* and *E*. The polarity at *D* is in opposition to that set up by the thermocouple *H*. If *G* is moved along the slide, a point will be found where the emf set up by the couple will be exactly counterbalanced by that set up by the battery between *D* and *G*. This point will be indicated on the galvanometer by showing no deflection.

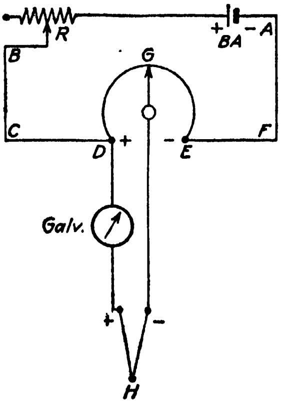

FIG. 25.—Simple potentiometer circuit.

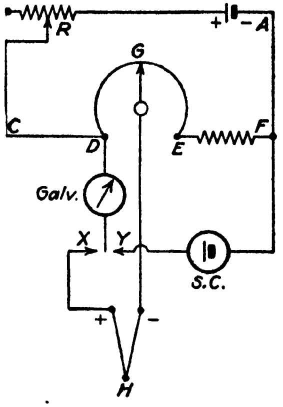

FIG. 26.—Potentiometer circuit with standard cell.

Modern potentiometers insert a standard constant-voltage cell with a fixed resistance in the circuit for standardizing the current in the slide wire. These instruments also often include manual or automatic cold-junction compensation.

Standardization is secured as follows: Contact is made at the point *Y* connecting the standard cell *SC* introduced in the modified galvanometer circuit as shown by Fig. 26. The rheostat *R* permits adjustment so that the current flow through the slide and the fixed resistance *EF* will produce an emf between *D* and *F*, balancing exactly that set up by the standard cell, as indicated by no deflection of the galvanometer needle. By this operation the current in the slide wire *DGE* has been standardized. By breaking contact *Y* and establishing contact *X*, the potentiometer is ready for use in measuring emfs generated by the thermocouple. To compensate for cold-end junction, the circuit is further complicated by the introduction of an additional sliding connection at *D* (Fig. 27). This permits varying of the emf between *D* and *G* by moving *D*, and compensating for the cold junction. The slider *D* is set to the known temperature of the reference junction, and the balance is established by slider *G* (as previously discussed), giving the true temperature of the hot junction. By certain other circuit changes, automatic variation for compensating for cold junction may be secured.

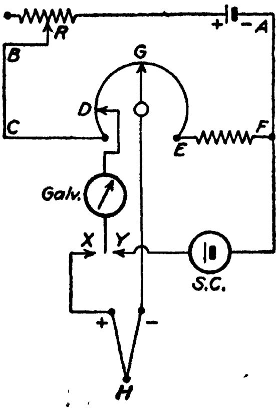

FIG. 27.—Potentiometer with cold-junction compensator. (*Leeds and Northrup.*)

Potentiometer type industrial *temperature recorders* and controllers, using thermocouples, are increasing in number and variety because of their advantage of high accuracy, the usual guarantee on such instruments being less than 1 per cent of full-scale deflection. These instruments are driven by synchronous motors and the timing, chart drive, standard cell check, and balancing may all be accomplished automatically by the same motor. In certain designs electron tubes or photoelectric cells are used to reduce lag and to eliminate mechanical parts and friction.

7. Radiation Pyrometers.—This pyrometer has a low-temperature limit of about 1000°F and an upper limit beyond the highest temperature demanded by the mechanical engineer. The instrument (Fig. 28) is based on the fact that the rate at which a black body radiates heat energy is a function of the temperature. It consists of a lens *B* mounted in a black tube. This lens collects and focuses the radiant heat energy from a given source on a sensitive thermocouple *C*. The emf thus developed is measured by a galvanometer or millivoltmeter connected at *D*.

When this device is sighted upon a large area, such as an incandescent fuel bed or a furnace wall, the temperature attained by the thermocouple will be a function of the temperature of the object sighted upon and will be almost independent of the distance from the object, as long as the entire field of the lens is covered. But the actual result will depend upon the emissivity of the source of radiation. The instruments are calibrated for black-body conditions and will be approximately correct when sighted into a large furnace through a small opening. The instrument readings may be 10 to 50 per cent low when sighted upon light-colored or metallic surfaces in the open. Radiation pyrometers are ordinarily used for temperatures above 1000°F, but by substituting a thermopile (series thermocouples) for a single couple the instrument may also be adapted for lower temperatures down to and including ordinary atmospheric temperatures. The radiation pyrometer is made in both indicating and recording types and is also used as the sensitive element for control devices.

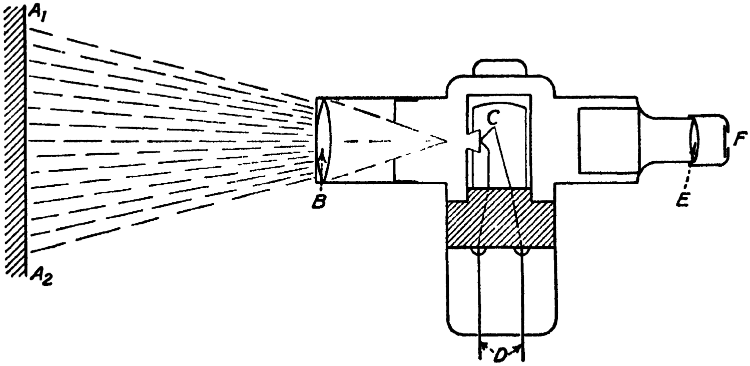

FIG. 28.—Diagram of a radiation pyrometer.

8. Optical Pyrometers.[1]—The rate of emission of luminous radiation from a body per unit area depends upon the brightness of the luminous area and upon the temperature.

One type of optical pyrometer compares the intensity of luminosity from a given surface with that from a source of light, the intensity of which is known. The standard is usually an electric lamp. The filament is heated (rheostat control) until its glow disappears against the background of the body the temperature of which is being measured (Fig. 29). If the relationship of current through lamp to temperature of filament is known, then the temperature for a given filament brightness may be determined and hence the temperature of the luminous body. Another type of optical pyrometer uses a standard light source of constant intensity. This source is then blocked off by a wedge of dark glass until its luminosity becomes equal to that of the surface under observation. The position of the glass wedge is calibrated in terms of temperature. These two types furnish the basis employed in the construction of most optical pyrometers. The optical pyrometer is well adapted for measuring the temperature of small luminous surfaces. The photoelectric pyrometer, which uses a photocell as the sensitive element, might also be classed as an optical pyrometer.

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[1] For further discussion see A.S.M.E. Code on Instruments and Apparatus, "Temperature Measurement," Chap. VIII, Optical Pyrometers.

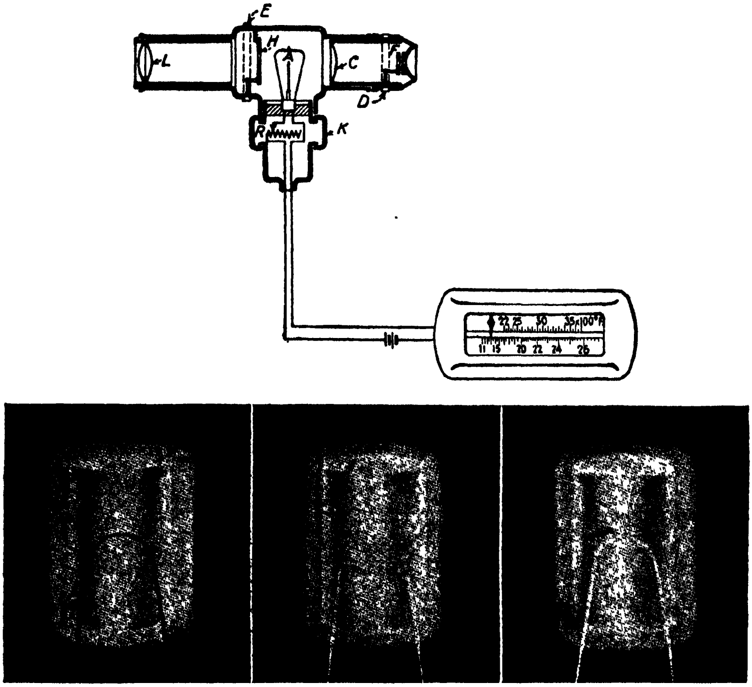

Fig. 29.—Schematic view of an optical pyrometer, and method of reading. (*Bacharach.*)

The accuracy of optical pyrometers under favorable conditions may be within 1 per cent. The optical pyrometer is used principally above the limit of base-metal thermocouples, but it may be used for temperatures as low as 1200°F. When black-body conditions do not prevail, the optical pyrometer is subject to considerable error, as is the radiation pyrometer. For molten metals in the open, for instance, the instrument may read as much as 300° low. Other sources of error are the presence of screens of smoke or gases and the necessity for visual setting which may introduce human error. In the determination of temperatures of luminous flames the optical pyrometer has been applied with some success by using color screens of two different wave lengths, red and green.

9. **Pyrometric cones** are a simple and inexpensive form of fusion pyrometer. The cones are small pyramids, about 2 in. high. Each cone is prepared from mixtures of oxides and glass, to give a definite melting point. A series of cones, with melting points 20 to 70° apart, covers the range from about 1100 to 3600°F.

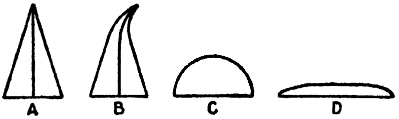

Fig. 30.—Pyrometric cones.

In using pyrometric cones, a series of four may be exposed in the furnace, and their behavior may be described as follows: *Initial deformation* is when the apex of the cone becomes rounded or has started to bend over (*B*, Fig. 30). *Softening temperature*, when the cone sags into a spherical shape (*C*, Fig. 30). *Fluid temperature*, when the cone becomes softened to the point of fluidity and spreads out as in *D* (Fig. 30).

The accuracy of temperature measurement with pyrometric cones cannot of course be closer than the interval between succeeding cones of the series. This interval is not uniform but is usually from 20 to 70°F. The behavior of the cones also depends upon the rate of heating and upon the furnace atmosphere.

EXPERIMENT 4. Calibration of Thermometer and Thermocouple Pyrometer

I. PREFACE

Since there is no direct or primary method of measuring temperature, there is no primary standard for calibrating temperature instruments. Temperature is measured only by its effect on the properties of materials, and the three properties most often used are: (1) change of state; (2) vapor pressure; (3) expansion. The first two afford means of calibration by reference to known boiling and melting points and saturation temperatures.

The common methods of calibrating thermometers and pyrometers may therefore be classed under three headings:

1. Comparison with Standard Instruments.—This is usually accomplished by the immersion of both instruments in a well-agitated bath of water, oil, or molten salt, or in an electric furnace or muffle. The comparison standard is usually either a mercury thermometer or a noble-metal thermocouple. Such instruments should have Bureau of Standards calibration certificates.

2. Checking by Reference to Known Boiling or Melting Points. The substances commonly used in this method are listed in Table VII.

TABLE VII.—FIXED POINTS FOR CHECKING TEMPERATURE-MEASURING INSTRUMENTS

	°F
Ice, melting at	32
Water, boiling at	212 (at 14.7 lb)
Naphthalene (cp), boiling at	425 (at 14.7 lb)
Tin, melting at	449
Lead, melting at	621
Zinc, melting at	787
Sulphur, boiling at	832 (at 14.7 lb)
Aluminum, 99.9 per cent melting at	1219
Sodium chloride (common salt), melting at	1473
Copper, melting at	1983

3. Checking by Reference to Saturated-steam Temperatures. This method is successful only when a dead-weight platform or other very accurate method of measuring the steam pressure is available (see Experiment 53, Pressure-temperature Relation for Saturated Steam, page 236). True temperatures are read from the steam tables (Table XLIII, page 473).

In this experiment the student will make a thermocouple, then calibrate it and use it. He will also adjust and operate the potentiometer instruments.

II. INSTRUCTIONS

1. Preparations.—Obtain a pair of copper-constantan wires (about No. 24 gage) from the instrument room. Secure also a standard thermometer, a thermos bottle, and thermocouple instruments of both the potentiometer and millivoltmeter types.

2. Setup.—The following equipment should be available: (*a*) Thermometer-comparison bath with mechanical stirrer. (*b*) Boiling-point apparatus, similar to that shown in Fig. 32. (*c*) Equipment for welding or soldering thermocouples. (*d*) Equipment for determining melting points (tin and lead melting points recommended).

3. Procedure.—The procedure of this experiment consists of four steps: (*a*) Remove the insulation from a ½-in. length at each end of both thermocouple wires. Make two copper-constantan junctions, *A* and *B* (Fig. 31), then cut the copper wire in the center and connect these two ends to the instrument at *C* (Fig. 31). (*b*) Place cold junction *B* in a thermos bottle filled with chipped ice and water. Attach hot junction *A* to standard thermometer and insert both in the thermometer-comparison bath. Take readings of both thermometer and thermocouple potentiometer over the range of temperature directed by instructor. Correct the thermometer readings for stem emergence by Fig. 17. (*c*) Determine the steam point at atmospheric pressure with the apparatus of Fig. 32 (or in violently boiling water). Read the barometer. (*d*) Determine melting points by inserting thermocouple into the molten metal in an iron crucible and taking readings every 15 sec during cooling. The freezing temperature will be indicated by a flat section on the time-temperature

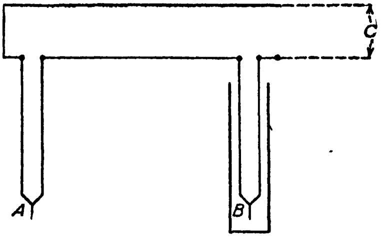

FIG. 31.—Thermocouple hot- and cold-junctions.

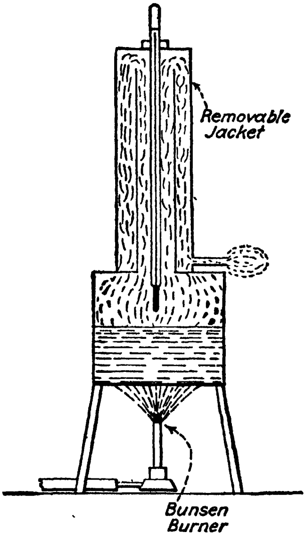

FIG. 32.—Apparatus for checking the boiling point of water.

curve. (Cover the molten metal with powdered charcoal to prevent oxidation.)

III. RESULTS

Plot calibration curves, with potentiometer reading in millivolts (or degrees) on abscissas, and true temperature on ordinates. On the same sheet plot the corresponding curve from Table VI. Report should include diagrams of connections.

IV. NOTES AND PRECAUTIONS

Accuracy of all temperature measurements depends on both instrument and method of application. Almost any required degree of accuracy in scale and calibration may be obtained with either glass-stem thermometers, electrical-resistance thermometers, or thermocouples, but large errors may be introduced by improper application. Most of these errors are due to: (1) Stratification and poor mixing of the fluid. (2) Radiation. (3) Incomplete immersion or conduction. As the difference between measured temperature and ambient temperature increases, these errors become greater. The following **precautions** should be observed in temperature measurements:

1. The filament in a mercury thermometer should be inspected for possible breakage and separation.

2. Thermometers should not be withdrawn for reading.

3. Thermometer wells should be of the proper type, with exposed parts insulated.

4. In measuring fluid temperatures, adequate stirring or mixing is important.

5. Thermocouple corrections for lead resistance and for cold-junction temperature should be applied.

6. When switches are used in thermocouple circuits, all junctions of dissimilar metals must be at the same temperature.

7. Fine-wire thermocouples give minimum time lag, high sensitiveness, and low radiation error.

8. Radiation errors may be reduced by covering the sensitive element with protecting shields or open-end tubes (see Fig. 33).

9. Errors by conduction of heat to or from the sensitive bulb should be guarded against.

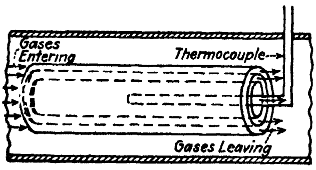

FIG. 33.—Radiation shields for thermocouple.

10. For surface temperatures, a fine-wire thermocouple is much more accurate than a bulb-type instrument.

Surface temperatures are best measured with thermocouples, but they may also be measured with thermometers or with radiation instruments. A thermometer pressed firmly against a hot surface and sealed with plastic material will read low, the error varying from about 5° when the true temperature is 50° above ambient, to 20° when the temperature is 200° above ambient. More accurate surface temperature measurements may be made with thermocouples, either attached to the surface with adhesive or imbedded in cement in a shallow groove. Fine wires (24 to 40) should be used, and the insulated lead wires should be in contact with the surface for some distance from the couple. For high temperatures or for quick readings from inaccessible surfaces, a radiation thermopile or a radiation pyrometer may be used.

MEASUREMENT OF WEIGHT

EXPERIMENT 5. Calibration of Portable Platform Scales

I. PREFACE

Determinations of the weight of materials or of the pressure created by certain forces are required for the solution of many engineering problems.

The direct method of weighing is more accurate than the computed weight from a volumetric determination. In direct weighing, the factors of temperature, expansion, density, etc., of fixed volumes are eliminated.

The method of direct weighing commonly employed is through advantage ratio. A series of standard weights hung on one end of the leverage system serves to balance an unknown weight suspended on the other end of the system. The given standard weight (producing equilibrium in the levers) multiplied by the leverage ratio gives the value of the unknown weight.

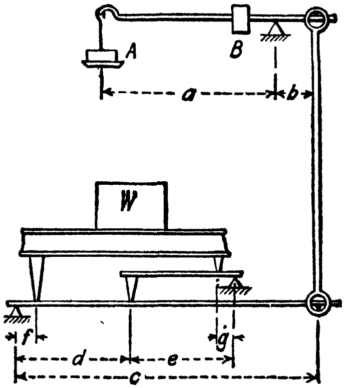

FIG. 34.—Leverage system of platform scales.

The form of machine incorporating the leverage system as described above is termed a "scale" (Fig. 34). Such scales when used for weighing very light objects are called "balances." Balances will not be further treated here, as the student previous to entering the mechanical laboratory has been instructed in their use in the chemical laboratory.

Scales have been constructed (testing machines) to weigh as much as 1,000,000 lb, while the chemical balance, such as is used in the determination of the proximate analysis of coal, the opposite extreme, should weigh to a minimum of one-ten-thousandth of a gram, or even less.

The common form of scale as used in most engineering determinations is termed a "platform scale." These scales with certain additional incorporated mechanisms may be made to dump automatically upon receipt of a definite weight. Other types, employing lever mechanisms, such as dial scales, are direct reading and are very convenient for use where rate of variation in weight is considered, such as in determining the gasoline consumption of an automobile engine.

A dump trap, which may be counterbalanced to dump a given weight, is often installed with a registering counter so as to determine rate of flow. Many intricate devices are to be found on weighing mechanisms which automatically establish equilibrium and in many cases record the weighing.

The deformation of a coiled or flat spring may be employed to give direct weighings. This type of weighing device is used in traction dynamometers, oil friction-testing machines, small "spring balances," etc. Any spring scale should be frequently calibrated by the use of standard weights.

The method of calibration of a platform scale is the same whether the equilibrium is established automatically or by manual control.

II. INSTRUCTIONS

Carefully level the scale so that the platform is in a horizontal plane. With no weight on the platform, adjust the balance lever so that the arm shall come to rest midway between the stops. This balance may be secured by adjusting the weight of the pan on the poise hook or by means of a special adjustable counterweight incorporated in the weighing mechanism.

By using standard weights and increasing the platform loads by equal increments, the error may be determined for a given range of loading. When reading the scale, the beam should be in the central position between stops.

Take the readings of the scale beam when a known standard weight is placed on each corner and in the center of the platform. Use the average of these weighings in computing error.

The **sensitiveness** of the scale may be determined for any loading by finding the mean weight that must be added and subtracted from the platform load in order to bring the balance arm from the middle position to the upper and lower stops.

Form 5.—Calibration of Platform Scale

General description

Type............................Laboratory No.............. ....
Maker....... ....Capacity...................., lb.
Beam graduations.................Leverage ratio..................

Platform load, lb.	Beam readings, lb.						Error, lb.	Error, per cent
	Corners				Center	Average		
	1	2	3	4				

Sensitiveness

Platform load, lb.	Beam reading, initial center load	Load increment to upper stop	Load decrement to lower stop
Zero..................			
One-fourth capacity....			
One-half capacity......			
Three-fourths capacity.			
Full capacity...........			

Date............... Observers:................................ ....
....
.. ..

Plot calibration curve between error of scale as ordinates and weight in pounds as read from scale beam as abscissas, plus errors plotted above and minus errors below the axis of abscissas.

Determine sensitiveness at no load, half load, and full-scale load.

Determine leverage ratio, load on platform to weight on poise hook.

III. NOTES AND PRECAUTIONS

1. Before and after using platform scales, always check the balance of the poise beam at zero load.
2. Be sure that each poise weight is of the proper ratio for the scale on which it is used.
3. Always check the full-scale position of the movable poise weight against a poise-hook weight of the same value. If they do not check, determine the reason and correct the difficulty before using the scale.
4. The error of spring scales due to change in altitude is about 1 part in 1,000 for 10,000 ft. There is no such error in lever-balancing or pendulum type scales.
5. Greater sensitiveness may be obtained by substituting "flexure plates" for knife-edges. In the Emery testing machine this construction gives an accuracy of 1 part in 1,000,000, while for a knife-edge unit of similar capacity the accuracy is about 1 in 30,000.
6. The tentative draft of the A.S.M.E. Code on "Measurement of Quantity of Materials," (Instruments and Apparatus, Part 5) gives a 50-page discussion of weighing scales.

MEASUREMENT OF VOLUME

Experiments 6 and 7

General.—Volumetric measurement is very closely related to direct weighing. All volumetric devices are calibrated initially by direct weighing. Where temperature variation enters, a set of calibration curves must be obtained for the volume device to compensate for the variation of specific gravity and volume.

The volumetric measuring devices listed in "Fluid Meters, Part 1" (A.S.M.E.) consist of the following: tank, piston, disk, rotary, bellows, and wet-drum meters, (see Chap. VII).

A **piston** or positive displacement meter or pump may be used to measure volume of flow. In this case curves of slip or ratio of delivery to displacement volume for various inlet and back pressures on the device must be obtained through calibration with a direct weighing mechanism. A series of such curves, together with the number of strokes per unit of time, serves to give the total volume passing in unit time. **Disk** and **rotary meters** of the positive displacement type are treated in like manner as the piston meter.

EXPERIMENT 6. Calibration of a Volume Tank

I. PREFACE

The **tank** is the most universally employed volume device. The volume of the tank per unit of height, if of constant cross section, furnishes the unit of measurement in determining the volume of its content for any given depth of filling. If accuracy is desired, where continuous (full tank) volume measurements are being made, the tank should be necked and an overflow apron provided so that the variation in volume for different tank fillings may be practically eliminated. Tanks of such construction (Fig. 35) are often used in the testing of large boiler units; they must be calibrated over the temperature range encountered in the test. Two tanks are usually necessary so that one may be emptied while the other is filling; but, where the rate of flow is uniform and continuous, a single tank is sometimes used, intermittent readings being taken.

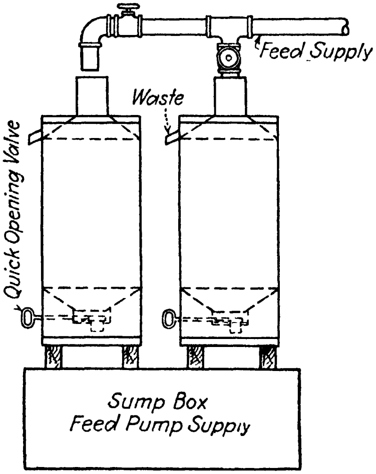

Fig. 35.—Volumetric water measuring tanks foi boiler tests.

Various types of gages and devices are used to indicate the volume content of tanks or receivers for different depths of the liquid. These indicators may embody, in their construction, floats, multiplying linkages, gas-compression chambers, gage-glass water columns, hook gages, etc. Some of the devices furnish the power to actuate indicators or recording pens which move along linear or circular scales.

For the part of a tank that has a constant cross section, the corresponding indicating scale graduations will be uniformly spaced, but for the other portions of the tank or for an irregular-shaped tank the spacings will vary.

For the calibration of a tank already fitted with a linear scale, it is merely necessary to establish the relationship between the reading on the scale and the volume of liquid in the tank. A standard volume measure may be used to introduce definite volumes into the tank; or the actual volume may be computed from the dimensions of the tank or determined from actual weighings and the consideration of density.

II. OBJECT

The object of this experiment is to calibrate a volume tank for various temperatures of contained liquid.

III. PREPARATIONS

1. Apparatus.—Volume tank, platform scales, thermometers, steel rule, and measuring tape.

2. Setup.—One method is to introduce a steam pipe through the upper opening of the tank and extend it down through the tank to very near the bottom. This pipe should have its bottom end capped and a series of holes drilled along the pipe extending for its entire submerged length. Some type of stirring device will be required to maintain a uniform temperature throughout the entire volume of water. If the tank is not necked but has its upper end full open, then a temporary loose cover should be used to prevent loss of heat.

IV. INSTRUCTIONS

Determinations.—Steam should be introduced into the water to heat the volume of liquid through a range of temperatures to about 180°. For accuracy, calibrated platform scales should be used. Correction for the displacement volume of both steam pipe and stirrer should be made.

The calibration should cover several water temperatures but need not proceed by exactly equal increments of temperature rise, as all points should fall on a smooth curve. Water must always stand at exactly the same level, when readings are taken.

Results.—Plot curve between the weight of water in the tank when filled (corrected for displacement of steam pipe and stirrer, and error of platform scale) as ordinates, and temperature of water in tank as abscissas. Plot similar curve to the above computed from tank dimensions, and water volumes and weights as determined from the steam tables.

Explain the cause of any variations between the two curves. Give reasons favoring either method. Discuss in full the accuracy of the temperature readings secured by the method used.

EXPERIMENT 7. Calibration of Bellows Gas Meter

I. PREFACE

Bellows gas meters are generally used in connection with the sale of fuel gases, but they are also suited to other measurements of clean gases or air in the temperature range from 32 to 110°. Such meters consist of a casing divided into two chambers by a vertical partition. Within each chamber is a measuring receiver in the form of a leather-walled diaphragm. Slide valves admit gas alternately into the flexible measuring receivers, and the reciprocating movements of filling and emptying actuate a set of counter dials. Connections are such that the movement of one bellows is at mid-position when the other is passing the end or dead point. Bellows meters are rated at a pressure drop of 0.5 in. of water and are usually adjusted to within ±1 per cent error, though the error may increase with use and be larger at the greater rates of flow. These meters are well suited for intermittent duty, and their accuracy is almost unaffected by variations in the flow rate. They may be obtained with sheet-metal casings for low pressures and with cast-iron casings for high-pressure lines.

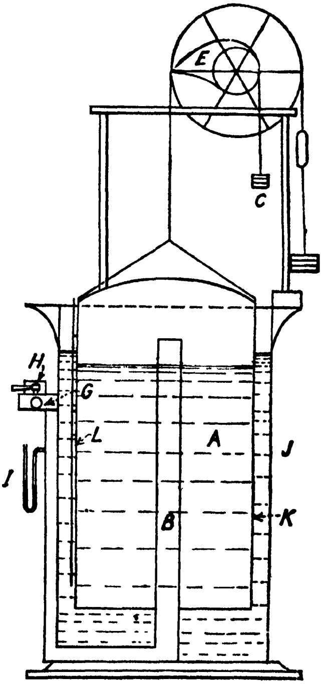

FIG. 36.—Gasometer meter prover.

The **gasometer** or meter prover is a standard apparatus which has supposedly been calibrated by means of a cubic foot bottle (certified by Bureau of Standards) (Fig. 36), and it is used for calibration of the bellows type of meter. This apparatus consists of two cylindrical tanks, each with one open end, of dimensions such that the smaller will have a diameter 3 to 6 in. less than the larger. The balancing mechanism, to compensate for the weight of the tank, consists of a weight *D* to balance in whole or in part the weight of the upper tank. The weight *C* compensates for the depth of immersion or buoyant effect of the bell or upper tank, hence the special design of cycloidal cam giving a variable effective radius.

II. INSTRUCTIONS

If the gasometer is to be used to calibrate a bellows meter, attach meter at *G*, the upper manometer cock. Use orifices on the outlet side of meter to secure different rates of flow at the proper differential pressure. Open cock *H*, and flow through the meter will start. Manometer pressure may be regulated by changing total weight *D*. Readings of meter are taken on the fly.

Readings of gas temperature and pressure within the bell, and a means of measurement of vertical linear displacement of the tank are required. A thermometer, introduced as shown, measures the temperature. A manometer *I*, as connected, serves to determine the pressure within the bell. A scale *L*, reading in cubic feet and decimals, is fixed vertically to the moving tank. A stationary indicating finger on the fixed tank serves to indicate the vertical distance moved or volume displaced.

The water in the holder and the incoming gas should be close to room temperature; otherwise, the charge should be held in prover until adjustment of gas temperature is made.

By use of these readings and the calibration curve of the prover, a calibration curve of the bellows meter may be obtained.

Plot curve between error of bellows meter in percentage as ordinates, and bellows-meter reading in cubic feet as abscissas, plus errors plotted above axis of abscissas, minus errors plotted below axis of abscissas.

Form your own blank for tabulation.

For a general treatment of the subject of meters for gases, liquids, and vapors, see Chap. VII, Measurement of Fluid Flow, page 263.

III. NOTES

If the meter prover itself is to be calibrated, the water, air, bottle, and prover should be at the same temperature. Also, the air in the prover must be saturated, as contact with the water in the standard bottle would absorb moisture and introduce a volume error.

The early method of determining capacities was based upon the standard English burner which consumed 6 cu ft of gas per hour, so a "5-light" orifice would pass 5 times 6 or 30 cu ft of gas per hour.

Capacities are now based on the volume of gas (sp gr 0.64) which will pass through the meter when the difference in pressure between the inlet and the outlet is ½ in. water pressure.

The domestic meter does not affect the moisture content of the gas being metered, as does the wet-drum meter.

Domestic meters (the tinned-sheet-steel type) operate on 6 to 8 in. water pressure. The meters, however, should preserve their accuracy at lower pressures to cover periods in which low pressures prevail, as in city gas mains.

The capacity of a meter increases with increase of pressure. Doubling the pressure increases the capacity about 1.4 times. However, it is poor practice to increase the capacity by this means; preferably, the next size of meter should be used.

If the diaphragm skins stretch, shrink, dry out, or leak, the metering will be inaccurate.

Meters constructed of special materials are procurable to meter butane, propane, hydrogen, nitrogen, oxygen, acetylene, etc. Iron-body meters may be purchased for almost any rated capacities and at pressures up to 500 lb or more.

Gasometer tanks similar to this meter prover but larger in size are used by engineers for various types of gas measurement where a high degree of accuracy is required, as in the calibration of flow meters (see Chap. VII, page 263) and for measuring the air supply to internal-combustion engines (see Chap. XI, page 455).

EXPERIMENT 8. Measurements of Speed and Time

I. OBJECT

The object of this experiment is (*a*) to make a brief study of the various types of speed-measuring instruments; (*b*) to illustrate the reasons for and the magnitude of the common errors in the measurement of rotational speed; (*c*) to calibrate a centrifugal tachometer.

II. PREFACE

The most frequent occasion for **time measurements** by the mechanical engineer is in connection with the determination of linear or rotational speed. The ordinary spring-wound clock or stop watch is the most common instrument, but, with the present accurate frequency control on electric utility systems, synchronous motor timing devices are becoming more and more common. System frequency is usually held so close to 60 cycles per second that synchronous timers are sufficiently accurate for all engineering uses. (Such accuracy should not be assumed on a small system without investigation.) If spring-driven timers are used, they should be calibrated over a period of several hours by comparison with a synchronous clock operating from a closely controlled electrical system.

High-speed studies may require more refined methods of timing than the ordinary stop watch or electric clock. The oscillograph and the motion-picture camera are rapidly becoming everyday tools for the engineer in high-speed analysis. By both of these instruments the moment-to-moment conditions in a rapidly changing system can be studied in detail. For measuring very short time intervals or for measuring longer intervals very accurately, both mechanical and electrical devices are available. The most common mechanical device is the tuning fork, usually arranged to draw a sine curve. By using forks of different frequencies, almost any desired subdivision of the second may be obtained. Chronographs may employ electromagnets to vibrate a pen or pencil, or the paper may be punctured at regular intervals by a high-tension electric spark produced by a mechanical interrupter. Alternating-current "cycle counters" have recently been introduced, the ultimate division in this case being a single a-c cycle.

Starting and stopping errors are the chief inaccuracies in timing speed measurements. Wherever possible the instruments should be mechanically or electrically connected so that a single impulse such as the throwing of a switch or lever will start both the timing and the displacement or counting instrument. Long runs are also desirable. An error of a single scale division on a $\frac{1}{5}$-sec stop watch amounts to 1.33 per cent error in a 15-sec run but only 0.11 per cent error in a 3-min run.

Rotational speed measurements are the most important engineering measurements of speed. When linear speed is desired, it can usually be obtained by simple calculation from rotational or angular speed. The three common methods of measuring speed are by (1) **revolution counter and timepiece**; (2) **tachometer**; (3) **stroboscope.**

The difference between a counter and a tachometer is aptly illustrated by the two parts of an ordinary automobile speedometer. In fact, the mechanism of a speedometer is almost identical with that of certain types of counters and tachometers. The tachometer is an instrument for indicating the angular speed of a rotating shaft in revolutions per minute (rpm) and corresponds to that part of the speedometer which indicates speed in miles per hour. The counter totals or counts the number of revolutions or of miles and tenths of miles, as the case may be.

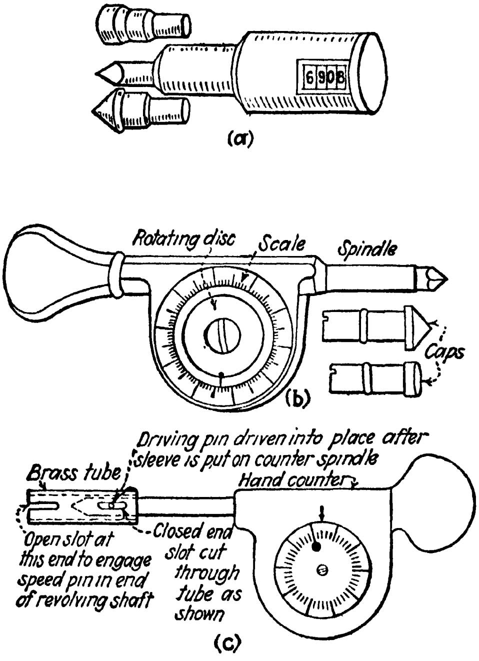
6908
(a)
Rotating disc
Scale
Spindle
Caps
(b)
Driving pin driven into place after sleeve is put on counter spindle
Brass tube
Hand counter
Open slot at this end to engage speed pin in end of revolving shaft
Closed end slot cut through tube as shown
(c)

FIG. 37.—Hand counters.

1. The Counter.—The hand counter (usually called "revolution counter" or "speed counter") consists of a light shaft or spindle geared to one or more counting dials. Perhaps the most common are the worm-and-wheel type (Fig. 37*b*) and the cyclometer type (Fig. 37*a*). In either case the end of the driving spindle is provided, as shown, with a conical rubber or a triangular steel-pointed tip. The usual method of driving the counter is to apply this rubber or steel tip directly to the drilled center on the end of the shaft, relying on friction to drive the spindle at the same speed at which the shaft is rotating. Some method of positive connection is preferred for speeds above 600 rpm. Well-made counters are reliable over a range of 200 to 2,000 rpm. Speeds as high as 5,000 rpm can be measured, but an accuracy greater than 1 per cent cannot be depended upon. The time of application of the counter is observed by means of either an ordinary watch, with a second hand, or a stop watch. From this reading, together with the initial and final readings of the dial of the counter, the speed is calculated.

The stop watch and counter are sometimes built in the same unit and connected in such a way that both are started and stopped simultaneously, or the counter may rotate continuously when applied to the shaft and its interconnection to the watch be controlled by a small lever or button. This double-unit arrangement largely eliminates errors in timing, and such a combined instrument is usually regarded as **the most accurate instrument available for speed measurements,** though its accuracy depends largely on that of the timing mechanism. The "tachoscope," the "speed indicator," and the "chronometric tachometer" embody the combined counter and timing mechanism.

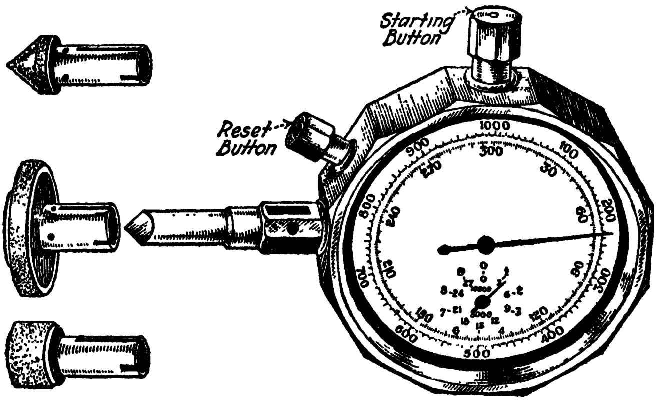

FIG. 38.—Hand speed indicator.

The **tachoscope** consists of a revolution counter and a stop watch mounted side by side in a frame. The two instruments are interconnected so as to start and stop simultaneously, but the duration of reading depends on the operator, and the instruments must be separately reset after each reading.

The **speed indicator** (Fig. 38) operates on the same principle as the tachoscope but averages the speed over a short period of time and indicates directly the speed in revolutions per minute. A single button winds and starts the watch, connects the drive shaft to the counting train, and automatically disconnects the counter after a definite period of time. Two instruments are available. On the low-speed instrument the large hand turns around once for each 100 rpm, and on the high-speed instrument it turns once for each 1,000 rpm. The errors of this instrument do not normally exceed 0.3 to.1 per cent for upper scale readings.

With the **chronometric tachometer** the operator presses a button to start the timing mechanism, but the disengagement and speed indication are automatic and the duration of reading is only 1 sec. The 1-sec readings are automatically repeated by the instrument as long as the counting and timing mechanisms are engaged, hence the justification for calling the instrument a "tachometer."

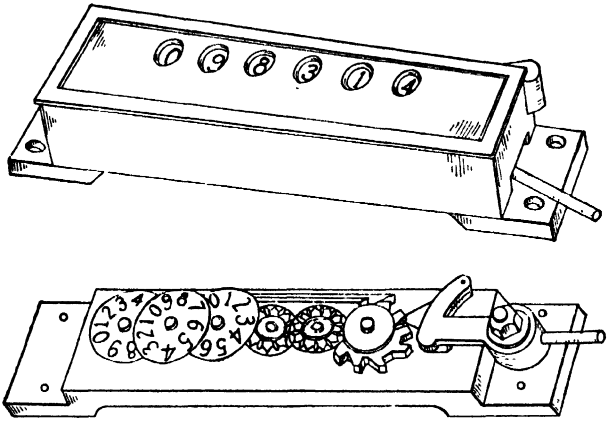

Fig. 39.—Stroke counter.

A rigidly mounted and positively driven counter (direct connected or geared) is preferred over the hand counter or portable type for accurate work. A counter so mounted is called a "continuous counter." When a controlled-frequency alternating-current electric circuit is available, the continuous counter with electromagnetic clutch and synchronous motor timer makes an excellent combination. Synchronous-timing motors reading to 0.1 sec may be obtained, and the two devices should be operated by the same switch.

In low-speed work (300 rpm or less), a reciprocating lever and ratchet drive is often used to actuate the counter mechanism, this lever being connected to any part of the machine from which reciprocating motion may be obtained. Such a counter (Fig. 39) is often called a "stroke counter." In cases where the stroke counter is applicable, it is usually operated continuously during a test, and, next to the direct-connected or geared counter, it furnishes the most accurate means of measuring average speed over a long period of time. Another type of counter (Fig. 40) is actuated by a pin or projection from a rotating member which turns the counter arm and operates the ratchet.

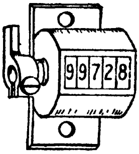

Fig. 40.

The A.S.M.E. Code prescribes the use of a continuous counter in engine, compressor, and pump tests. For test work involving short runs, the continuous counter with electromagnetic clutch furnishes a most convenient and accurate speed-measuring device. One of the most familiar applications is in connection with electric dynamometers (see Figs. 82 and 83, pages 104 and 105).

2. The Tachometer.—The tachometer gives a direct and continuous indication of speed and is therefore the most convenient instrument for observing speed variations or fluctuations and for general observations in which a high degree of accuracy is unnecessary. It is often made recording and applied to such machines as turbogenerators, conveyors, paper machines, gas engines, etc., for purposes of control and record of performance.

The actuating mechanism of the more common tachometers is either: (1) a centrifugal device similar in construction to a centrifugal flyball governor (Fig. 41); (2) an electric generator or magneto (Fig. 42); (3) a centrifugal fan or pump; (4) a vibrating reed. The centrifugal type is usually the least expensive and the easiest to use in test work because of light weight and portability. Good makes are likely to have an error of 15 to 30 rpm between 600 and 2,500 rpm, when new. After much use there may be a 50- to 100-rpm error. The electromagnetic instruments are much used, particularly in connection with the testing of electric machinery where observations of instantaneous speed variation are important. The main advantages of this type are distant location of the indicator and good

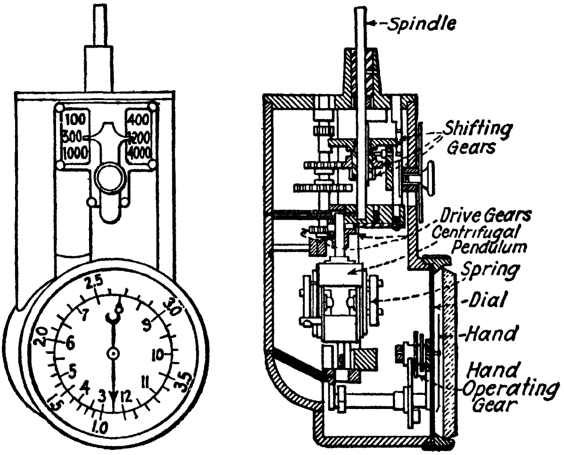

FIG. 41.—Centrifugal hand tachometer.

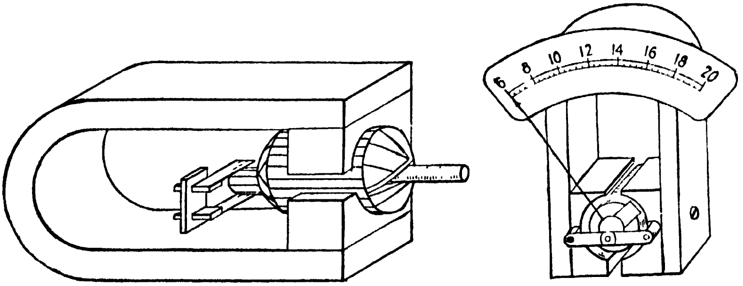

FIG. 42.—Electric tachometer (generator and indicator).

accuracy over a wide range. The fan or pneumatic type is a simple and sturdy unit and is easily made recording. The same may be said of the liquid or centrifugal-pump type. The vibrating-reed tachometer has the disadvantage of having a small range; but, for permanent application to installed machines, a wide range is of course unnecessary. These tachometers are self-contained and have the advantage of requiring no connection with the rotating shaft. They are merely bolted or held firmly in place against the machine under test. The accuracy of this tachometer is within 50 to 100 rpm.

Certain other types of tachometers are occasionally used (see A.S.M.E. Code). For electric alternators, a frequency meter may be calibrated in revolutions per minute.

Practically all tachometers of the above described types have two main disadvantages: (1) their readings are affected by certain external conditions such as temperature, atmospheric pressure, magnetic fields, and mechanical vibration; (2) their indications are subject to a time-lag or inertia effect, which means that their indications do not respond so quickly as might be desired to changes in speed. Friction, wear, or age also affects the accuracy of some types. These errors are usually so small that an occasional calibration will give results sufficiently accurate for ordinary work. Certain precautions are to be observed, however, in using tachometers, as indicated in Sec. V, page 70.

3. The Stroboscope.—The stroboscope utilizes the phenomenon of persistence of vision when an object is viewed intermittently. This is also the basis of the motion picture. By viewing a cyclic motion at the same point in the cycle each time, the object appears to be motionless. By changing the frequency slightly, slow motion can be obtained. The older stroboscopes interrupted the vision either by a tuning-fork arrangement or by a rotating perforated disk or cylinder. The neon-tube stroboscope is now displacing the other types, because of its convenience and because it is adapted to stroboscopic photography.

Neon-tube stroboscopes are available for speed measurement, with indicating dials calibrated throughout the range from 700 to 14,000 rpm. These instruments are especially valuable where it is inconvenient to make a connection or contact with the rotating shaft, or for light-powered machinery where the load to drive a speed-measuring instrument would affect the operation of the machine. Although the stroboscope is used for speed measurements, its main application is for studying machine motions and vibrations, especially in high-speed internal-combustion engines, textile machinery, and the like.

Speed measurements by the stroboscope necessitate its calibration in terms of frequency. The frequency of a neon-tube stroboscope is conveniently checked against the frequency of an a-c power system. In fact the common neon-glow lamp obtainable at retail stores for a few cents may be operated from any ordinary utility power source for illuminating a stroboscopic disk. This apparatus can then be used for calibrating tachometers at the 60-cycle synchronous speeds, 3,600, 1,800, 1,200 rpm, etc. A satisfactory stroboscopic disk may be made from a 3-in. black disk bearing a single radial white line of some width. At 3,600 rpm this disk will show two lines; at 1,800 rpm it will show four lines, at 1,200 rpm it will show six lines.

III. APPARATUS

1. Instruments.—The following portable instruments should be obtained: Two or more hand counters (including Starrett and Veeder), a tachometer, stop watch, and an ordinary watch.

2. Setup.—The apparatus for this experiment consists of two positively driven shafts operating at the same speed and so arranged that counter or tachometer may be applied to any or all of the four shaft ends. Means for varying the speed, either electrically or mechanically, is provided. The calibration standard is applied to one of the shaft ends. This standard may be either:

1. A positively driven continuous counter timed by a calibrated stop watch. The combined instrument (tachoscope) may be used to good advantage here.
2. A continuous counter with electromagnetic clutch and timed by a synchronous motor reading to 0.1 sec.
3. A stroboscope disk illuminated by two neon-glow lamps.

If method 2 or 3 is used, the electrical source must be a controlled-frequency alternating-current circuit suitable for precision operation of electric clocks.

IV. INSTRUCTIONS

One of the objects of this experiment is to demonstrate the difficulties, both personal and instrumental, in the way of obtaining really accurate observations of angular speed or revolutions per minute. To do this in minimum time, four observers are necessary, each using a different instrument. The following procedure is suggested:

Observer 1.—Read standard instrument (one of the three already mentioned) at time intervals called by observer 3.

Observer 2.—Read centrifugal tachometer at time intervals called by observer 3.

Observer 3.—Read Veeder counter, using stop watch and calling time every 5 sec.

Observer 4.—Read Starrett counter, using ordinary watch.

The foreman or chief of the group should act as observer 3. Readings should be made at five or more different speeds from the lowest to the highest available. Four runs are to be made at each speed—two runs of 2 min duration each, and two runs of 15 sec duration each. The tachometer observers will record their observations every 5 sec and should attempt to determine for each interval the mean rather than the instantaneous reading. Data sheets should be prepared before the observations are begun. The "average revolutions per minute" by tachometers, as recorded on the final data sheet, should be the average of the 5-sec readings.

A number of preliminary runs should be made. No data should be recorded until the group finds it possible to make two successive duplicate runs that check each other closely. When the teamwork has been developed to this stage, all preliminary readings should be discarded, and the test begun.

Section V, Precautions, should be read before any of the observations are begun.

3. Curves.—The following curves should be plotted, using instrument readings as abscissas: (1) readings by Starrett counter, against per cent error in these readings; (2) readings by Veeder counter, against per cent error; (3) readings by centrifugal tachometer against per cent error; (4) readings by centrifugal tachometer against error in revolutions per minute. All readings should be in revolutions per minute, and the average revolutions per minute as shown by the standard is to be used as the basis in all cases.

In plotting these curves, the vertical scales representing error should be so chosen that the smallest unit read on the instrument is represented by approximately the smallest division on the graph paper. Both the 120-sec and the 15-sec values should be plotted on the same graph, using different symbols for the two sets of points. Curve 4 is the calibration curve for the centrifugal tachometer as it would probably be plotted for actual use.

4. Report.—It is intended that the purposes of this experiment shall be brought out by the curves called for above. The discussion of results should accordingly cover the following items: (1) effect of method of timing on accuracy of speed readings with the counter; (2) effect of duration of readings on accuracy of speed readings with the counter; (3) effect of speed of shaft on accuracy of speed readings with the counter; (4) advantages and disadvantages of each type of counter; (5) accuracy of tachometer calibrations; (6) accuracy of the particular tachometer used in the experiment.

V. PRECAUTIONS TO BE OBSERVED IN USING SPEED-MEASURING INSTRUMENTS

1. All speed-measuring instruments depending on a friction tip for drive must be held parallel to the shaft under observation and firmly (but not too heavily) against it. This is very important, as the accuracy of readings with even the best instruments depends on a positive drive.
2. When making speed observations with a counter, the timing should, for most accurate results, always be carried out by the observer holding the counter. When one observer times another, an additional step is introduced into the process, and the time lag is greater and less uniform.
3. When speeds above 600 rpm are to be measured, a positive drive is recommended (Fig. 37).
4. When a speed counter is used for accurately measuring linear speed of belting, lathe work, etc., by the use of a rubber-tired "surface wheel," care must be taken to measure the wheel diameter accurately, with allowance for compression of the tire.
5. When there is any uncertainty as to which speed range of a tachometer is to be used, the highest range should always be tried first.
6. The change gears on tachometers should not be operated while the spindle is rotating.
7. When a continuous counter is used to obtain total revolutions during a long test, it should be watched carefully, for, if vibration causes it to shift position slightly, it may cease to indicate. Moreover, certain of these counters sometimes give trouble from "sticking."
8. When a "tachoscope" is used, the observer must take care that *all* hands are set to zero before each new reading. On some instruments this resetting requires three operations.
9. In using the vibration tachometer, the disturbing effects due to the vibration of other machines than the one being indicated must be guarded against, when placing the instrument.
10. The stroboscope can be successfully used in determining the revolutions of light-powered machinery where the load to drive another type of indicator would materially reduce the speed.
11. It must be remembered that speed-measuring instruments contain delicate mechanisms which may easily be broken by rough handling.

MEASUREMENT OF AREA

EXPERIMENT 9. Examination and Use of Planimeters

I. PREFACE

The simplest method of measuring areas is by means of the foot rule and the application of geometric formulas (see Fig. 57).

For very large regular or irregular areas, surveying instruments are used to determine the dimensions and outline of the area. A plot may then be made to such a scale as to make possible a drawing-board analysis of the plot surveyed. This area may be evaluated by means of instruments. In many engineering operations, areas of small dimensions are generated which may be evaluated or integrated automatically, by the use of hand instruments called "planimeters."

Polar Planimeter.—The polar planimeter is built in many different forms. The Amsler type is the one most commonly used, owing no doubt to its adaptability to general work.

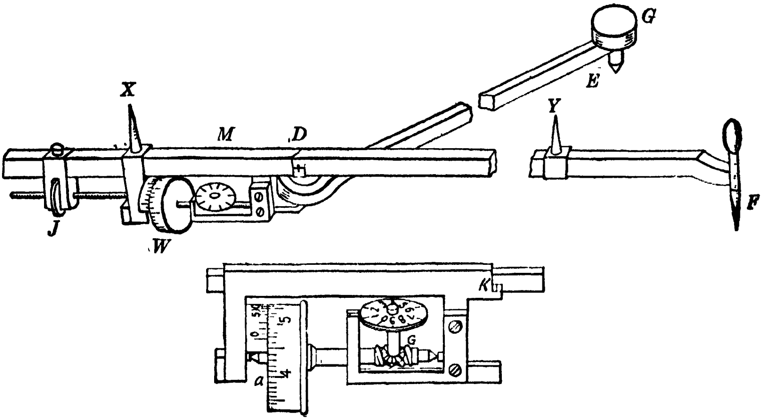

FIG. 43.—Adjustable polar planimeter.

A polar planimeter is shown in Fig. 43. It consists of two arms *ME* and *MF* pivoted at *M*. The arm *EM* during use is free to rotate around the point *E* which is held in place under the weight *G*. The arm *MF* carries at one end a tracing point *F* which is traced around the border of the area to be measured. On the extension of the arm *MF* is carried a wheel *W* whose axis is in the same vertical plane as the arm. The rim of the wheel *W* is in contact with the paper, and any motion of the arm *MF* except in the direction of its axis will cause the wheel to revolve; a graduated scale with a vernier denotes the amount of travel of the circumference. The circumference of the wheel is divided into 100 equal parts which are subdivided by a vernier into tenths, so that the reading of the vernier is in terms of thousandths of a revolution.

To read the area by the adjustable polar planimeter: Accept the formula

$$A = LR, \tag{3}$$

where A = area corresponding to the proportional part of a complete revolution of the record wheel, square inches.
R = circumference traversed by the record wheel, inches.
L = length of the tracing arm, inches,
= DF or XY in Fig. 43.

Generalizing the above formula to apply to an area of any size inside the "zero circle," we may state that "the area of any figure traced with a polar planimeter is numerically equal to the product of the length of the tracing arm and the circumferential distance moved through by a point on the record wheel." If the measurements are made in inches, the product (area) will of course be in square inches. Since the circumference of the record wheel is divided into tenths and hundredths, and the vernier reads in tenths, a reading in decimals to three places may be made on the wheel in those units for which the arm is set.[1]

To read mean height directly by the use of the adjustable polar planimeter: Assume as before that $A = LR$ and $A = lp$, where p is the mean ordinate of the diagram and l the length.

$$LR = lp, \text{ whence } \frac{L}{l} = \frac{p}{R}. \tag{4}$$

Now, since L is adjustable, we can make it equal to the length of the diagram l. We shall then have p the length of the mean ordinate equal the reading of the record wheel multiplied by the proper factor.

Adjust the distance between the steel points on the arm MF to the length of the indicator diagram. The difference of the readings of record wheel before and after tracing the diagram multiplied by the ratio of the scale of spring used to 40 will give the mean height.

NOTES.—The adjustable arm permits of reading areas in square decimeters, square inches, square feet, also of giving the average height of an indicator diagram in fortieths of an inch. The instrument demands extreme care in handling, as it is delicate and sensitive to shocks. Use a flat, smooth, unglazed surface for the record wheel to slide over. The most accurate results will be obtained when the arms of the instrument remain approximately at right angles to each other during tracing of a given diagram. Trace around diagram in the direction of the hands of a clock for positive readings.

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[1] For discussions covering the theory of the planimeter, see A.S.M.E. Power Test Codes, "Instruments and Apparatus, Part 15, Measurement of Surface Areas."

In the above discussion of the polar planimeter nothing was said of the **zero circle.** The zero circle is defined as the circle which will be generated by the tracing point when the arms of the instrument are clamped in such a position that the plane of the record wheel passes through the fixed center. It is evident that when the arms are so fixed, the tracing point may generate a complete circle with no movement of the record wheel. When the planimeter is used for areas larger than that of the zero circle, the area of this circle must be determined and added to the area indicated by the record wheel to obtain the total area enclosed by the movement of the tracing point.

Test rules or standard circular plates may be obtained from the manufacturers of planimeters for the purpose of conveniently checking these instruments.

Coffin's Averaging Planimeter.—This is a special form of the Amsler type in which the end of the arm slides in a straight slot instead of developing a circular arc. Area evaluations are not dependent upon a particular position of the diagram for tracing.

The Coffin planimeter has the following merits: (1) the quick setup of instrument and diagram permits of great speed in evaluating areas; (2) the convenience of having in connection with the instrument a record board, clips for holding cards and at times guiding tracing point, and a smooth hard surface for recording wheel to travel upon; (3) simplicity of the instrument, freedom from pivoted joints, and but one rigid member.

Wallis Planimeter.—In this instrument the shaft of the recording wheel is at right angles to the tracer bar instead of being parallel with it. Hence the wheel does not slide (as in the previous type described) but has a motion entirely rotative; this eliminates to a very great extent the surface roughness of the traced diagram.

This planimeter is provided with a scale to correspond with every indicator spring in common use, affording a means of reading the mean height directly.

II. INSTRUCTIONS

Rule up a 2-in. square and check area, with polar and Coffin planimeters.

Check formula (3) by measuring the parts of the planimeter and tracing a 5- by 2-in. rectangle with the instrument adjusted to read in square inches.

Determine experimentally the effect on the planimeter reading of a "negative loop" on a steam-engine indicator diagram, such as shown by Figs. 65 and 66, page 96.

Since L is adjustable it can be made equal to the length of diagram l. Formula (4) then reduces to $p = R$, or the length of the mean ordinate will equal the reading of the record wheel multiplied by the proper factor. Test by ruling up a 2-in. square.

Secure from the instrument room a standard circular plate or a "test rule" for checking planimeter. Determine the error of the instrument by obtaining the average of several trials.

Find area of indicator card (Fig. 64, page 96) by use of polar planimeter and Coffin's averaging instrument.

Determine the area of the zero circle for the polar planimeter.

EXPERIMENT 10. Use of Integrator for Circular Charts

INSTRUCTIONS

The integrator obtains the average radius of records traced on circular charts of uniform graduations. Directions for operating the Bristol-Durand type are as follows: Use the special graphs or conversion charts which are furnished with each instrument for converting the readings of the integrating wheel to the units of the particular chart being averaged. A separate curve is necessary to fit the units of each circular chart. The chart to be averaged should be placed over the stationary socket in the center of the wooden base, and the rotatable pin should be inserted in this socket. The shaft extension of the integrating wheel should be secured in place by the set screw in the hub of the wheel, and the triangular support should be slipped on and secured to the shaft so that the vernier will stand in a vertical position when the shaft is dropped into the vertical slot of the rotatable pin. Before beginning to trace a record, the zero on the integrating wheel should be made to coincide with the zero on the vernier, and the direction of the tracing point over the record should be such that the reading on the integrating wheel will be from zero upward.

In tracing a record that does not cover a complete revolution of the chart, the tracing point should be started at one end of the record line. After the complete length of the record to be averaged has been traced (whether the record covers part or a complete revolution of the chart), the tracing point should be returned to a point on the chart having the same radius as the starting point, the movement of the tracing point being along a line corresponding or parallel to one of the time arcs.

To obtain the average radius for a complete record of the entire circumference, the reading may be referred directly to the corresponding graph accompanying the instrument. To obtain the average radius where the record is less than a complete revolution of the chart, the reading on the integrating wheel must be divided by the number and fraction thereof of the time spaces covered by the record, and multiplied by the total number of these time spaces on the whole circumference of the chart. Take the figure thus obtained and transfer to the graph to obtain the average radius in terms of chart units.

THE INDICATOR AND ITS USE

EXPERIMENT 11. Examination of Indicators

I. PREFACE

The indicator is an instrument that makes a graphical record of the variable relation existing between the pressure in a cylinder at any time and the corresponding position of the piston in its stroke. If this record is made for both the forward and back stroke, a closed cycle will be obtained.

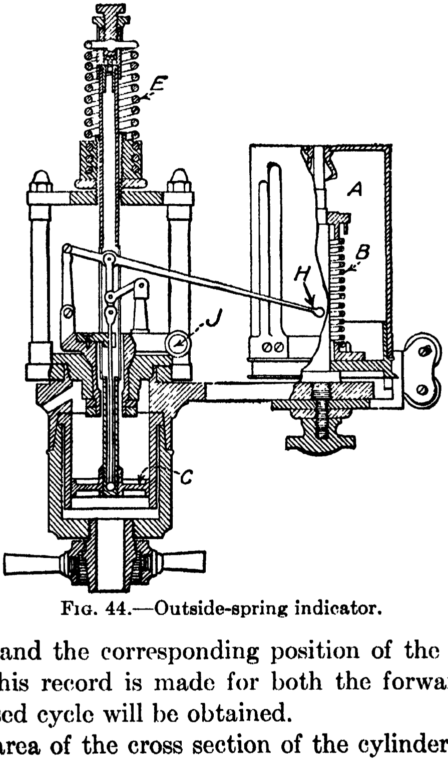

FIG. 44.—Outside-spring indicator.

Since the area of the cross section of the cylinder is constant, the horizontal coordinate of the graph or diagram which proportionately locates the position of the piston in its stroke at any given time may be scaled to represent either piston displacement or volume. The indicator therefore records the variable pressure-volume relation of the cylinder gas for the entire length of both strokes if the diagram be properly scaled.

Figure 44 shows an outside-spring type of indicator. *A* is the drum around which the diagram card is fastened by means of clips attached to drum. This drum is given a forward motion by the pulling of the cord which is attached to the crosshead of the engine (or to any other reciprocating member, the movement of which is proportional to that of the crosshead) through some type of reducing motion. The return motion is secured through the uncoiling of the spiral spring *B*.

The piston *C* is actuated by the pressure in the cylinder, the movement of this piston being opposed by the spring *E*. The movement of the indicator piston, and hence of the marking point, is thus proportional to the pressure in the engine cylinder (this pressure may be either above or below that of the atmosphere). By means of the handle *J*, the marking point *H* may be pressed against or withdrawn from the drum.

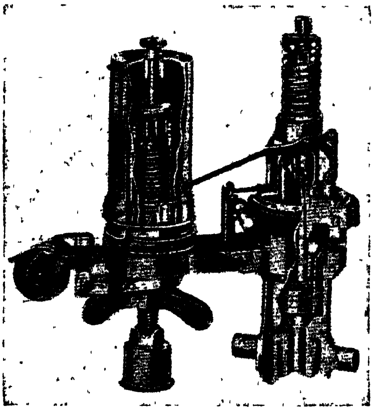

FIG. 45.—Sectional view of Maihak standard indicator.

Figure 45 presents a rugged type of indicator, so constructed as to permit interchange of cylinders and pistons. This indicator may be supplied with an integrator attachment which automatically planimeters the diagram of every engine cycle and totalizes these values during the entire test run, independent of whether or not the diagram is actually drawn.

The selection of an indicator should be controlled by the factors of engine speed, maximum cylinder pressure, and rapidity of pressure change. For slow-speed engines the inertia effect of the moving parts back of the pencil motion is not sufficient to force the recording point beyond its true position. However, for high-speed engine indication the inertia effect is multiplied, and provision must be made to annul the inertia effect to secure true indication. The inertia effect may be minimized by using stiffer springs, making operating parts lighter and reducing drum travel.

Some makes use a large piston with a very stiff spring, thereby reducing the weight of the recording-lever mechanism per unit of piston area.[1]

Indicators are usually made in three sizes, giving cards approximately 5 × 2¾ in., 3½ × 2 in., and 2½ × 1¼ in., and used respectively for low-, intermediate-, and high-speed work.

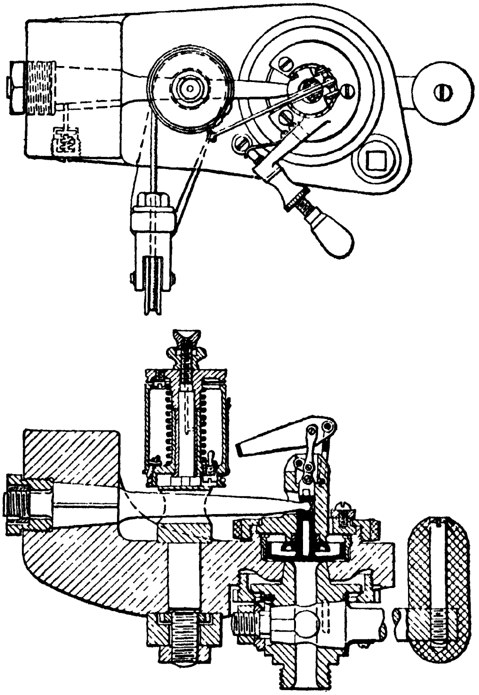

Fig. 46.—Plan and sectional view of Maihak high-speed indicator.

Figure 46 presents the internal construction of a special design of indicator for high-speed (2,400 rpm) and high-pressure (600 lb) use. This indicator, however, is not limited to high-speed engines; it applies also to medium- and moderately high-speed engines where pressure changes occur abruptly during some portion of the engine cycle, although the rotative speed of the engine may be comparatively moderate, as in certain types of solid-injection diesel engines with explosive combustion, or in uniflow steam engines with auxiliary exhaust valves provided for the automatic control of compression.

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[1] For an exhaustive study of the development and use of indicators of all types see "The Engine Indicator," K. L. De Juhasz, *Instruments*, June, 1932, to July, 1933 (14 chaps.).

The introduction of a cantilever spring has a twofold advantage: First, it supplies ample force for actuating the pressure-recording mechanism so that the force of friction between the writing stylus and the paper is negligibly small in comparison. Second, owing to its high period of natural vibration, it gives extreme rigidity to the recording system so that the writing stylus follows the rapid variations in pressure without any discernible vibrations.

For *high-speed engine research* in the internal-combustion engine field there are several modifications of the mechanical high-speed indicator. **Microindicators** trace very small diagrams that must be greatly magnified for analysis. **Optical indicators** obtain magnification by various types of optical systems, and the diagram is usually recorded by photographic means. There are at least 10 well-developed models of the optical or light-beam indicator, most of which have been originated and used principally in European laboratories. Optical indicators consist essentially of a piston or a diaphragm in the pressure chamber, operating against a beam spring and transmitting a very small motion to a mirror which reflects light from a point source. A motion is imparted to the mirror in another plane (or to a second mirror in the system) by some type of connection to the engine piston or shaft. The card drawn by the light beam is magnified by the optical arrangement and can be either viewed on a ground-glass screen or photographed.

The **electrical indicator** is a recent development in which the characteristics of an electric circuit are made to change by the changes in pressure in the cylinder. Either the resistance, capacity, impedance, or potential in the electrical circuit may be varied by the pressure element. The electrical potential method utilizes a piezoelectric crystal instead of an indicator spring as a pressure-responsive element. With a vacuum-tube amplifier and a cathode-ray oscillograph, synchronized pressure-time curves may be obtained. Electrical indicators have the advantage of distant reading.

A number of **multicycle indicators** have been developed, in which an indicator diagram is built up point by point from a large number of engine cycles. The Bureau of Standards indicator is of this type.

Several forms of *maximum-pressure indicators* are available for showing only the maximum pressure in an internal-combustion engine cycle. A simple form uses a balancing spring to oppose the cylinder pressure. A contact and a neon-light flasher indicate the point of balance. The adjustment is by a micrometer screw calibrated in pounds per square inch.

II. INSTRUCTIONS

1. Procedure.—Examine various types of indicators as furnished by the instructor.

2. Precautions.—The pencil mechanism is built light in weight so as to cut down the inertia effects. Handle with care—do not force or put undue pressure on any part. Do not remove pins or screws from the joints of mechanisms.

3. Determinations.—Put in and remove spring in each type, noting how to secure adjustment of tracing point for use with pressures below atmosphere.

Note whether provision is made for different sizes of pistons. Record piston diameters.

Make record of the pressure springs and their scales accompanying the instrument.

Make sketch of straight-line mechanism.

Determine ratio of motion of pencil point to that of piston.

Determine maximum throw of pencil point.

Determine maximum length of card that may be secured.

III. NOTES

The object of the linkages is to secure a straight-line motion and also to magnify the vertical movement of piston. This permits the use of a stiff spring with small displacement.

The relation between pressure per square inch and actual movement of pencil point fixes the scale of spring. A ½-in. area of piston is usually taken as the basis for determining the spring scale. An X-lb spring is one that allows the marking point to deflect 1 in. when the ½-in.-area piston is subjected to a change in pressure of X lb per sq in.

In high-speed work, stiff springs are essential on account of the inertia of the moving parts.

High-speed operation with long engine-indicator connections may cause lag by throttling or the condensation of steam; it is good practice to use two indicators in such cases, making the connections as short as possible. For this work a single straight-way valve of plug or gate type may be used.

The tension of the drum spring should be checked to see that no lag or jump is present. This lag may produce a distorted diagram.

When the pencil point is made of brass and the coated cards are used, a sharp, well-defined diagram is obtained and a minimum of pencil friction is present.

A nonstretchable indicator drum cord should be used and it should operate in a line parallel with the piston rod.

In preparing the indicator for use, the piston should be removed and lubricated with cylinder oil. If the piston is free when placed in the cylinder, it should drop of its own weight.

After the indicator is adjusted on the engine, the engine should be turned over and the indicator drum checked for clearance from striking against stops.

The above adjustments and precautions should be carefully made as errors from 5 to 10 per cent in diagram area may result from lack of their consideration.

On account of the corroding effect of ammonia gas, brass cannot be employed in the construction of indicators for use on ammonia compressors. Monel metal or stainless steel is used. If the piston and cylinder of the brass type of indicator are well oiled, a limited number of diagrams may be secured without seriously affecting the metal. After such use the indicator should be thoroughly washed.

EXPERIMENT 12. Indicator Reducing Motions

I. OBJECT

The object of this experiment is to calibrate an indicator reducing motion.

II. PREFACE

A steam-engine indicator cannot in general be operated directly from the crosshead, for the stroke of the engine is many times longer than the circumference of the indicator drum. Reduction mechanisms must therefore be utilized. These devices are subject to errors in design and errors due to wear in joints and should be checked for inaccuracies.

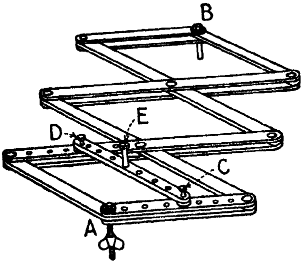


FIG. 47.—Universal pantograph reducing motion.

Many different types of indicator reducing motions are to be found in use. A number of the more simple types are shown in Figs. 47 to 53. Figure 47 shows the universal pantograph, and Fig. 48 represents its setup in practice. A modified four-linkage type of pantograph is shown in Fig. 49.

Figure 50 gives the setup for a Brumbo type of reducing motion. In the type of Fig. 51, the crosshead link is replaced by a slot in the pendulum arm. Reducing mechanisms for direct attachment to indicators and capable of reduction for 6-ft strokes or less are to be found in practice. These are usually called "reducing wheels" (Fig. 52).

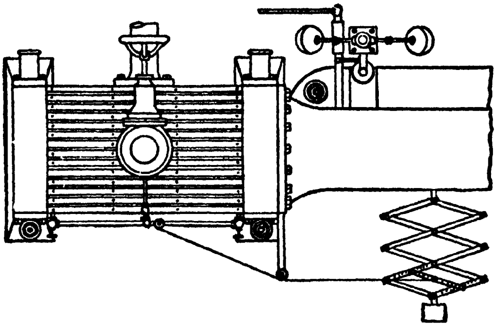


FIG. 48.—Pantograph in service (plan view).

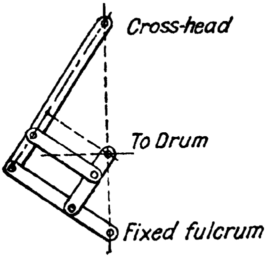


FIG. 49.—Four-bar pantograph.

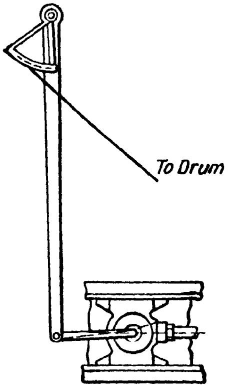


FIG. 50.—Brumbo reducing motion.

By using the proper reducing motion, the indicator-drum motion may be taken from any part of the engine which has the same relative reciprocating motion as the engine piston. The crosshead is generally used.

Figure 53 shows an ingenious method for indicator hookup. *A* is the rotating crankshaft of engine. *B* is a pin set in the shaft, off center, to furnish a crank arm for connecting rod *C* which has its other end pivoted to sliding member *D* by means of pin. From this sliding member, motion is given to the indicator cord,

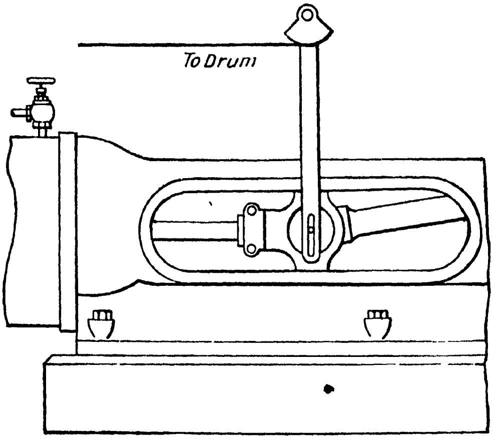


FIG. 51.—Slotted-pendulum reducing motion.

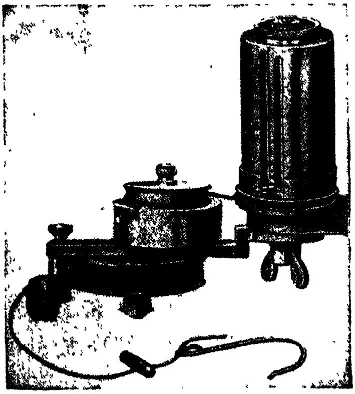


FIG. 52.—Reducing wheel attached to indicator.

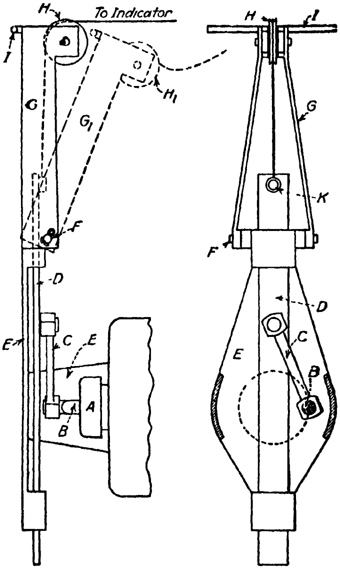


FIG. 53.—Crank-type reducing motion.

as presented in cut. Member *G* is made rotatable on pin *F* so that when in the vertical position the indicator cord is drawn taut and the indicator put into operation. By rotating member *G* by handle *I* to the position as shown by the dotted lines the indicator cord is put in a sag condition. There need be no hooking of cords with this device. It furnishes a very satisfactory method for medium-high-speed engines.

III. PREPARATION

Affix indicator on engine, adjust properly to avoid striking stops, and remove piston spring.

IV. INSTRUCTIONS

1. Procedure.—Put engine on either dead-center position (see Experiment 23, page 130), and make vertical pencil record of this position on the indicator card by pressing recording point against card and then lifting pencil motion. Move the crosshead forward by successive accurately measured increments of say one-tenth the length of the stroke, until the opposite dead-center position is reached; make record at each of these positions by drawing a corresponding vertical line on the indicator card.

Repeat this process and obtain card for reverse direction of motion of crosshead.

2. Results.—If the distances between the lines on the indicator diagram are all equally spaced and exactly vertical, the motion is correct; otherwise, a distorted card will be drawn and corrections must be applied or the reducing motion repaired or corrected in design.

Plot a curve between engine-piston movement in inches and linear travel in inches of point on indicator drum.

Determine the reducing-mechanism ratio.

EXPERIMENT 13. Indicator Springs

I. OBJECT

The object of this experiment is to calibrate a steam-engine indicator spring for pressures above atmosphere.

II. PREFACE

Indicator springs as furnished by the makers of indicators are as a rule quite accurate. Owing, however, to wear in joints of pencil motion, piston clearance, temperature, etc., it will be found that the actual pressure in the indicator cylinder does not correspond to that represented on the scaled indicator card, and a calibration is required. Further, indicators are used under conditions of high- and low-temperature ranges which modify somewhat the spring calibration.

Indicator springs are usually double-coil spiral springs, calibrated in terms of the pressure required at the piston to produce a movement of 1 in. at the marking point. An indicator piston of ½ sq in. area is the normal or standard size, but other sizes are used, with the same springs. The resulting changes in spring scales are as follows:

TABLE VIII.—INDICATOR-SPRING SCALES

Piston size, sq in.	Indicator-spring scale											
1	4	6	8	10	15	20	25	30	40	50	60	75
½	8	12	16	20	30	40	50	60	80	100	120	150
⅕	20	30	40	50	75	100	125	150	200	250	300	375
⅒	40	60	80	100	150	200	250	300	400	500	600	750

There are two general methods of calibration, the cold and the hot, better known as the dead-weight and the steam-pressure methods. Good practice would approve the method that furnishes conditions most nearly like those under which the indicator is to be or has been used.

III. INSTRUCTIONS

1. Procedure for Cold Calibration.—For an outside-spring indicator hot calibration is not necessary because the spring is not subjected to high temperatures in service. Inside-spring indicators, when they are to be used for indicating air or ammonia compressors or water pumps may also be calibrated cold.

The indicator is attached to an ordinary dead-weight gage tester see Fig. 14, page 27, and Experiment 2, page 26, for instructions). After placing a card on the indicator drum, erect a vertical line through the point given by the pencil position for both the extreme and stop positions of the drum. Use pencil and straightedge. With decreasing pressures during the calibration, draw lines from the right-hand vertical line toward the left-hand edge of the card, and for increasing pressures draw the lines from the left-hand vertical line toward the right-hand edge of the card. The indicator pencil point is pressed against the drum and the latter rotated to give a graphical record of the pressure as transmitted through the indicator spring.

For each reading from zero to maximum, lightly press the recording pencil arm downward with the finger and let it come to rest, so that any initial friction will act to hold it down. When the highest load has been reached, carry the pencil mechanism up slightly higher, then bring it down to balance by lightly tapping the cylinder with a small piece of wood. This should be done at each decreasing pressure.

For vacuum calibrations the vacuum-gage calibration apparatus shown in Fig. 16, page 31, is to be used, the mercury manometer readings being converted into pounds per square inch by calculation.

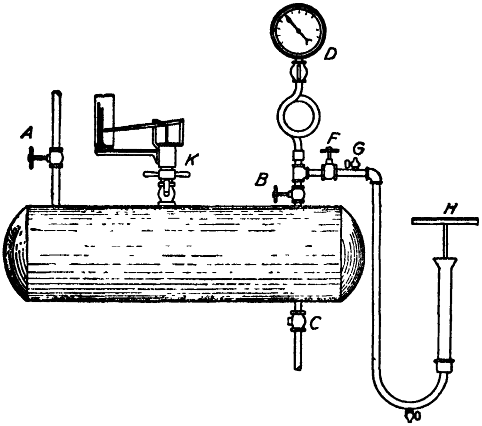


Fig. 54.—Steam-drum apparatus.

2. Procedure for Hot Calibration.—Hot calibration is usually used when an inside-spring indicator is to be applied to a steam cylinder. A steam-drum apparatus is used, as shown in Fig. 54. The internal pressure in the drum is registered on the indicator *K*, the gage *D*, and the dead-weight unit *H*. The gage merely serves to show the approximate pressure, the exact pressure being obtained by balancing the dead weights. The procedure for marking the card is the same as for cold calibration, given above.

3. Determinations.—Calibrate spring by increments as assigned by instructor. A sample card is shown in Fig. 55.

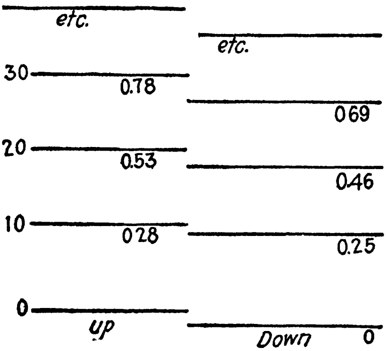


FIG. 55.—Indicator-spring calibration diagram.

Care should be used not to permit the pressure to fall during a rising calibration or to rise during a falling calibration.

4. Results.—The actual pressure as registered by the dead-weight tester, minus the pressure determined from the scaling or measuring of the calibration card, represents the error in pounds per square inch at the pressure under consideration. This error divided by the scaled pressure will give the error of the spring at this pressure.

To determine the mean spring scale for any load, divide the gage pressure on piston by the mean ordinate of the up and down readings as scaled from the indicator card. Use this value as the true spring scale from zero to the load selected.

THE INDICATOR CARD

Experiments 14 and 15

PREFACE

In the preceding description of the indicator it was noted that the pencil point moved in an exact vertical line in step with the indicator piston and that the linear distance moved through by a point on the drum was in exact proportion to the engine-piston displacement. This compound movement recorded by the pencil point will give a closed diagram which for a steam engine will be similar to that of Fig. 56, *A* representing conditions on one side and *B* the condition on the other side of the piston. Two indicators

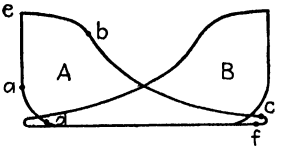


FIG. 56.—Typical steam-engine indicator card.

are sometimes used, one recording the pressure-volume relation for the head end and the other recording this relation for the crank end of the cylinder.

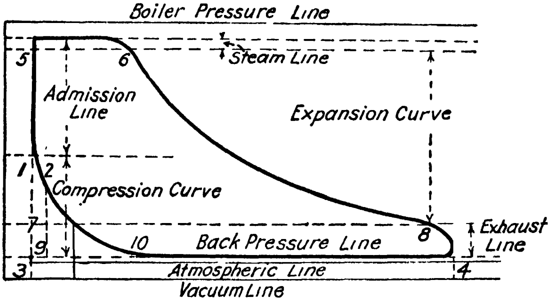


FIG. 57.—Analysis of indicator diagram.

The events in the stroke and phases for the complete cycle are designated in Fig. 57 and are defined as follows:

Point of admission (2) is the point on the indicator diagram that indicates the beginning of admission through the steam port. In percentage of stroke it is

$$\frac{4-2}{4-3} = \frac{\text{horizontal projected distance measured between 4 and 2}}{\text{length of card}}.$$

Point of cutoff (6) is the point on the indicator diagram that indicates the cutoff (closing) of admission steam port. In percentage of stroke it is $\frac{6-5}{4-3}$. The actual point of cutoff is determined by looking up along the expansion curve and spotting the point where the reverse curve begins, the point of inflexion. (For commercial point of cutoff, see page 89.)

Point of release (8) is the point on the indicator diagram that indicates the start of opening of the exhaust port. In percentage of stroke it is $\frac{7-8}{3-4}$. It is located by looking down along the expansion line and spotting the point where the reverse curve begins, point (8).

Point of exhaust closure or compression (10) is the point on the indicator diagram that indicates the closure of the exhaust port or the beginning of compression. In percentage of stroke it is $\frac{4-10}{4-3}$.

The atmospheric line is drawn on the original card when the engine is in motion. The three-way cock is shut off from the engine cylinder and opened to atmospheric pressure. The pencil point will then draw the horizontal straight line that represents the atmospheric pressure level.

EXPERIMENT 14. Geometry of the Indicator Diagram

I. INSTRUCTIONS

Constructions on Actual Indicator Diagrams.—Obtain an indicator diagram from one of the laboratory engines, and make on this card the following geometrical constructions. (Be sure that this diagram includes an atmospheric line, and record also the main steam-line pressure and the barometer readings.)

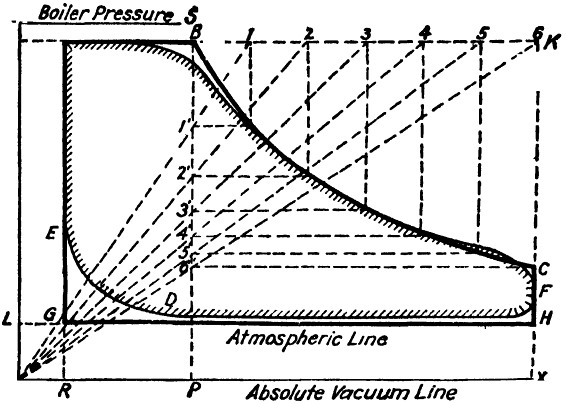


FIG. 58.—Actual and ideal indicator diagrams.

To draw reference boiler-pressure line YS (Fig. 58), read the steam gage pressure corresponding to that in the main steam line, and find length of the ordinate *LY* representing this pressure when reduced to the spring scale used; then draw a line parallel to the atmospheric line.

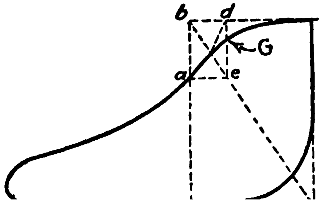


FIG. 59.—Determination of point of commercial cutoff.

To draw the reference zero vacuum line OX, read the barometer and convert the reading in inches of mercury to pounds per square inch.

Having determined the scale of the card, measure down from the atmospheric line to a point representing the given absolute pressure corrected to the spring scale, and draw the horizontal vacuum line.

To draw the reference clearance line OY, knowing the volumetric clearance of the engine: From the point *R* where the admission line cuts the vacuum line, lay off the clearance distance *RO* (to the proper scale) on the extension of line *XR*, and at *O* erect the clearance line *OY* perpendicular to *OX*.

To determine the point of commercial cutoff (Fig. 59): Through a point *c*, showing the maximum pressure during admission, draw a line parallel to the atmospheric line. Take a point *a* along the expansion line after the point of actual cutoff has been completed and preferably near the point of cutoff, and erect through *a* a perpendicular *ab* to the vacuum or atmospheric line and extend until it cuts the pressure line *cb* at *b*, which line is drawn through point of maximum pressure during admission. Through *b* draw *bf*, *f* being at the intersection of clearance and vacuum lines. Where *bf* intersects *ae* at *e*, erect a perpendicular *ed* to the boiler-pressure line. Then *G* is considered the point of commercial cutoff.

Where the steam line is irregular, as may be in case of the high-speed engine, take the mean of the vibrations at the highest part of the curve to establish the line *bc*.

Construction of the Ideal Diagram.—Construct an ideal indicator diagram corresponding to the actual diagram on which the previous constructions were made. The procedure is as follows:

Draw lines *OX* and *OY* at right angles as shown in Fig 58 Choose convenient scales to represent pressure and volume, then lay off *OR* on *OX* equal to the percentage of volumetric clearance of the engine, and *RX* equal to its length of stroke. Locate points on the pressure axis representing the atmospheric, boiler, and initial pressures, and draw the horizontal lines *LH*, *YS*, and *BK* through these points. Locate the point of cutoff *B* according to the assumed ratio of expansion. Since the expansion line *BC* is assumed to follow the law PV = a constant, it must represent an equilateral hyperbola. For the construction of the equilateral hyperbola, the following method may be used.

Let *B* (Fig. 58) be a point on the curve for which (P_1V_1), the pressure and volume, are known. Through *B* draw parallels to *OX* and *OY* as shown From *O* draw a diagonal intersecting *BP* at 1′ and *BK* at 1. Complete the rectangle with 1–1′ as a diagonal. The fourth corner of the rectangle will then be a point on the curve. From the construction above, prove by geometry that $PV = P_1V_1$. Other points may be determined from a repetition of this process, as shown in the figure.

EXPERIMENT 15. The Logarithmic Indicator Card

I. PREFACE

Determination of the Exponent n.—The value *n* in the equation $PV^n = C$ is dependent upon the type of prime mover, its loading and general operating conditions as to piston and valve clearances, condensation losses, leakages, power medium, and types of expansion.

The logarithmic or Clayton diagram may be used in the determination of the value *n*. This diagram may be constructed by plotting the P,V (absolute values) relations as determined from an indicator card on logarithmic (loglog) coordinate paper (see Fig. 60.

It will be found that the expansion and compression curves on the PV diagram will plot as straight lines on logarithmic paper.

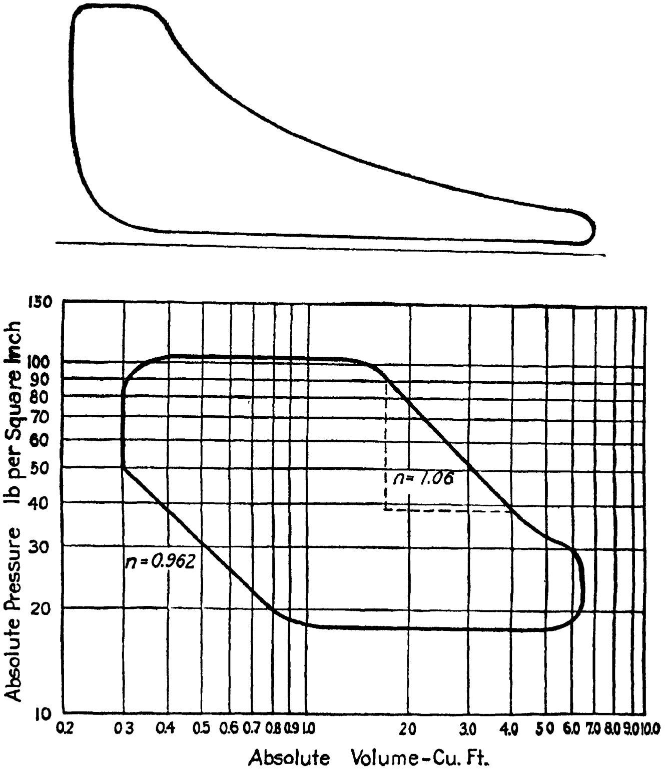


Fig. 60.—Logarithmic indicator diagram from Corliss engine.

There are a number of formulas that assume the form required for an analysis by use of logarithmic paper.

See Experiment 19, friction test of belts:

$$T_1 = T_2 e^{\theta f}$$

Flow over weirs (page 290):

$$Q = Ch^n$$

Expansion or compression of gases:

$$P = CV^{-n}$$

It is demonstrated in the Appendix under the heading Solution of Equations by the Use of Logarithmic Paper that the general equation of a straight line (logarithmic rectangular coordinates) will be of the form $Y = AX^{\pm n}$. If the expansion or compression curves of an indicator card taken from a gas compressor, steam or gas engine, be properly scaled and transferred to logarithmic paper, they will be found to plot as a straight line (see Fig. 60). We may therefore express the law of expansion or compression of gases by the formula $Y = AX^{\pm n}$.

Replacing by the corresponding symbols for gases, we have

$$P = CV^{\pm n}, \text{ or } PV^{\pm n} = C.$$

It will be found that the various types of expansion and compression curves usually met with in practice when plotted on log paper will give values of n between 1.0 and 1.4.

II. NOTES

If the plot on log paper of actual experimental values determined from an expansion be a curve, then n will not be a constant but will vary for each successive approximate straight-line section of this curve.

In any particular case, however, n may be determined for any given section along the expansion curve by considering the section a straight-line element. For a closer value, find the average n for a number of straight-line elements comprising the section under consideration.

The reason for the value of n changing during an expansion and deviating from a straight line (log coordinates) is that either the character of the expansion is changing continuously (if a smooth curve is developed) or that the character or the quantity of the substance itself is changing, as for example in the case of vapors, where either moisture or superheat may change during the process of expansion. The expansion may be such as to pass from a superheated region through a dry saturated condition and finally into a wet-vapor state.

III. INSTRUCTIONS

Secure a number of indicator diagrams from various machines, and by the use of log paper (or logarithmic coordinates on ordinary graph paper) determine the values of n, as already described.

Record the fixed dimensions of each machine, piston clearances, stroke, bore, and piston-rod diameter.

If volumetric clearances have not been determined, use the following method:

To draw the reference clearance line OY, actual linear clearance not known:

First method—using the compression line (Fig. 61). Draw a straight line across the compression curve intersecting the line of zero pressure OX at C. Measure from point a on the extension of this line the distance ad equal to cb and draw YO perpendicular to OX through d. Then TB divided by AT will be the percentage of linear clearance.

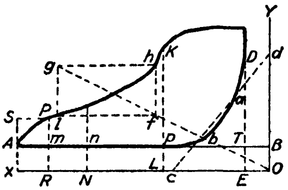


Fig. 61.—Determination of clearance from indicator diagram.

Second method—using the expansion line (Fig. 61). Construct the rectangle $lfhg$ and draw a diagonal gf and produce to intersect the line XO. This will give the point O and by erecting a perpendicular to XO we obtain the clearance line OY. Fair results are obtained by using the above graphical method when a clearly defined and undistorted expansion curve exists, of sufficient extent to admit of the application of the process. Clearances obtained by construction are, however, to be considered approximations.

Common Cross-section-paper Analysis.—Plot the expansion line of an indicator diagram on common cross-section paper or draw cross-section lines on the original diagram and determine the average value of n for several sections along the expansion and compression lines by the application of the following method.

$PV^n = C$ represents the equation of condition for a perfect gas for a point having the coordinates P and V. For the particular points (P_1V_1) and (P_2V_2), we have the equations

$$P_1V_1^n = C = P_2V_2^n$$

from which by taking logs we obtain

$$n = \frac{\log \{P_1/P_2\}}{\log \{V_2/V_1\}}.$$

This equation means that the value of n so determined by substitution for the points P_1V_1 and P_2V_2 located on any expansion or compression line gives the average value of n for that length of section.

EXPERIMENT 16. Calculations from the Indicator Diagram

I. PREFACE

Indicator diagrams may be used to make the following determinations:

1. Mean effective pressure (mep) or mean pressure of steam acting against the piston for the entire length of stroke.
2. Approximation of the weight of steam used in a complete cycle.
3. The valve action, *i.e.*, indicating the points and times of opening and closing of steam passages to and from the cylinder as well as in indicating wire drawing through ports and passages and distribution of steam between the two ends of the cylinder.
4. Distribution of developed power among the cylinders in a multicylinder engine.

II. INSTRUCTIONS

Secure indicator cards from both high-pressure and low-pressure cylinders of one of the compound engines in the laboratory (or from a simple engine if a compound is not available). Make the analysis called for by Form 16. Be sure to take all the necessary measurements from the engine, including those required for computing the clearance. The steam-consumption figures will not be available unless a test of the engine is made. See Fig. 57, page 87, for definition of terms.

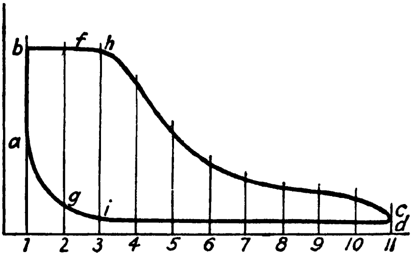


Fig. 62.—Determination of mean effective pressure by the ordinate method.

Determination of Mean Effective Pressure (mep) from Actual Diagram. 1. *Method by Ordinates.*—Divide the length of the card as represented in Fig. 62 into 10 equal spaces, and through their successive points draw the ordinates as indicated. Adding the length of ordinates inside the closed figure as measured in inches and dividing by the number of ordinates considered will give the length of the mean ordinate, and this, multiplied by the scale of the spring, will give the mean effective pressure. The larger the number of ordinates the more accurate the determination of the mean effective pessure.

2. *Method by Planimeter.*—The usual method of finding the mean effective pressure is to measure the area of the indicator card by means of the planimeter and divide this area by the length of the card. This quotient, multiplied by the scale of the spring, gives the mean effective pressure. By adjusting the polar planimeter to the length of the card the mean effective pressure may be read directly from the instrument (see Experiment 9 on the Use of Polar Planimeter, page 70).

Determination of the mean effective pressure for the ideal diagram can readily be made from the formula,

$$\text{Mean effective pressure} = \frac{P}{R}(1 + \log_e R) - p. \qquad (5)$$

P = absolute steam pressure at engine throttle, pounds per square inch

$R = \dfrac{\text{length card}}{\text{length to cutoff}}$ (theoretical card) = ratio of expansion.

p = absolute back pressure of exhaust, pounds per square inch.

Determination of Cylinder Horsepower.—The mean effective pressure determined as above multiplied by the area of the engine piston in square inches will give the force acting on one side of the piston. This product, multiplied by the piston travel in feet, will give the work done on one side of the piston per stroke.

The effective piston area is not the same for both ends of the cylinder, since on the crank end the area of the piston rod must be subtracted.

The travel of the piston in feet per minute for each end of the stroke is found by multiplying the length of the stroke by the revolutions of the crankshaft per minute.

If P_h = mean effective pressure for the head end,
P_c = mean effective pressure for the crank end,
A_h = head-end area of piston in square inches,
A_c = crank-end area of piston in square inches,
L = length of stroke in feet,
N = revolutions per minute of engine,

then the indicated or cylinder horsepower (ihp) will be:

$$\text{Indicated horsepower} = \begin{cases} \text{head end: } \dfrac{P_hLA_hN}{33{,}000}. \\ \text{crank end: } \dfrac{P_cLA_cN}{33{,}000}. \end{cases} \qquad (6)$$

The total indicated horsepower or cylinder horsepower of the engine is the sum of the indicated horsepower for the head end and the crank end.

The quantity $LA/33{,}000$ is called the **engine constant.** Time may be saved where a number of diagrams are to be considered by computing this quantity for each end of cylinder for use in the indicated-horsepower formulas.

Form 16.—Indicator-card Analysis

Name of engine................................Manufactured by ....
Type of engine { Vertical or horizontal Condensing or non-condensing
Type of engine { Simple or compound Uniflow or counterflow
Type of governing ....Type of valve............Usual r p m . ...
Horsepower rating....Diameter high-pressure cylinder....Diam. low-pressure cylinder...
Length of stroke........Diameter rod high-pressure....Diameter rod low-pressure ...
Barometer..................

Diagram Analysis

Item	High-pressure cylinder						Low-pressure cylinder					
	Head end			Crank end			Head end			Crank end		
	C.O	Rel	Com.	C.O.	Rel.	Com.	C.O	Rel	Com.	C.O	Rel.	Com.
Pressure (gage) at												
Pressure (absolute) at............												
Percentage of stroke at.....												
Area of card ...												
Length of card..												
M e p...........												
Clearance by computation.....												
Piston displacement.........												
Engine constant.												
Hp.............												
Steam per i.hp-hr. as weighed..												

Date..................... Observers:.....
....
....

III. NOTES

Diagram Factor.—If the engine be assumed to exhaust at atmospheric pressure, that is, is noncondensing, the ideal diagram (Fig. 58, page 88) will be bounded by the heavy lines *ABCHGA*. The actual diagram will vary from the ideal somewhat as shown by the shaded lines (see also high-speed card, Fig 63). The admission line will slant slightly downward and the point of cutoff will be rounded, owing to the slow closing of the valve. The first half of the expansion line will fall below the theoretical, owing to a drop in pressure caused by cylinder condensation, but will rise above the theoretical in the latter part of the stroke on account of re-evaporation due to heat given out by the hot cylinder walls to the low-pressure steam. Instead of the pressure dropping abruptly at *C*, release takes place just before the end of the stroke, and the diagram is rounded at *CF* instead of having a sharp corner. The back pressure line *FD* is drawn slightly above the atmospheric line on account of the pressure drop caused by the throttling

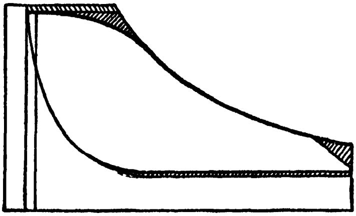


FIG. 63.—Comparison of actual and ideal diagrams.

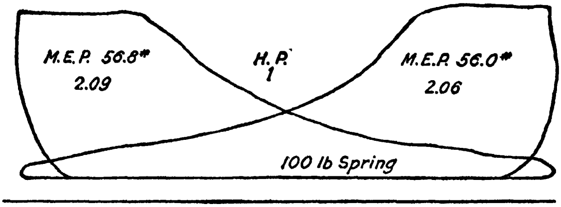


FIG. 64.

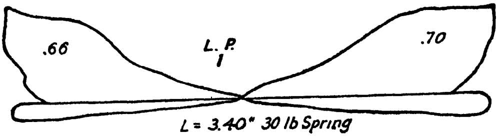


FIG. 65.

FIGS. 64 and 65.—Cards from compound Corliss, high- and low-pressure cylinders, operating noncondensing, light loads.

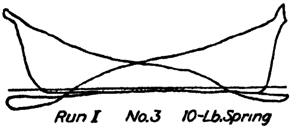


FIG. 66.

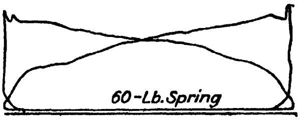


FIG. 67.

FIGS. 66 and 67.—Light-load and full-load cards, slow-speed throttling steam engine. Note spring scales.

of the exhaust steam through the ports. At D the exhaust valve closes and compression begins, rounding the bottom of the diagram up to E. *The ratio of the actual to the ideal mean effective pressure is termed the diagram factor e*

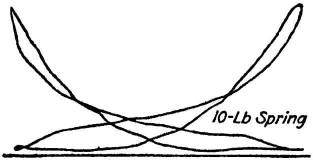


Fig. 68.

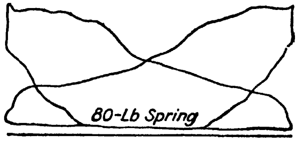


Fig. 69.

Figs. 68 and 69.—No-load and full-load cards for high-speed automatic cutoff engine.

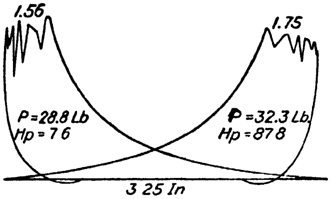


Fig. 70.—Straight-line steam-driven air compressor.

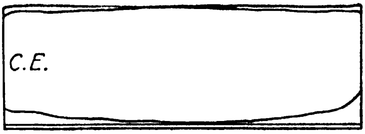


Fig. 71.—Steam pump, steam admitted during full stroke.

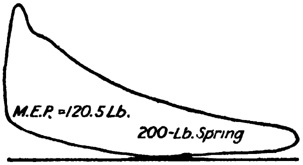


Fig. 72.—Gas engine, slow speed, four cycle, large displacement.

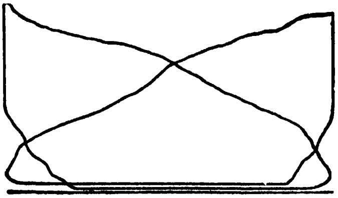


Fig. 73. Uniflow steam engine, full load.

In an engine producing a perfect or ideal diagram, it should be noted that no clearance is considered, no wire-drawn steam losses, perfect hyperbolic expansion, full boiler pressure as initial pressure, and terminal pressure that of the atmosphere for a noncondensing engine, and of the condenser for a condensing engine.

The diagram factor for large medium-speed engines e = 0.90.

For small, medium-speed engines, e = 0.85.

For high-speed engines, e = 0.75.

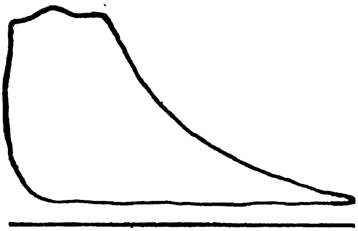


Fig. 74.—Carbon-dioxide compressor.

Typical cards from various machines are shown herewith.

Diagrams of water hammers, explosive pressures, valve-chest steam pressures, steam- or air-pipe pressures, etc., may also be obtained by use of the indicator and the variation or fluctuations in pressure recorded. The indicator in general use may register pressure against time, distance, or volume.

MEASUREMENT OF POWER

Experiment 17

I. PREFACE

The devices used for the measurement of power are classed under the general name of dynamometers. Dynamometers are of two kinds: (1) **absorption dynamometers,** which absorb the power, as it is being measured, and (2) **transmission dynamometers** which measure, with a very small absorption of power, the power that is being transmitted from one machine to another.

The most apparent application of the dynamometer is its use in the measurement of the power developed by the various types of prime movers, *i.e.*, engines, turbines, and motors. Designers and manufacturers of automobiles or of steam engines or of electric motors must be able to measure the output of their machines in order to supply purchasers with performance figures as well as to secure this information for the development of improvements and new designs.

The dynamometer makes it possible for the designer of the various types of both driven machines and prime movers to verify or disprove the theoretical assumptions that he has arrived at from the laws of mechanics, hydraulics, and thermodynamics. Again, the purchaser of a driven machine or prime mover may make use of a dynamometer to check the performance guaranteed by the manufacturer.

Railroads and locomotive manufacturers make extensive use of dynamometers, both on the road and in locomotive-testing plants. In addition to being used in the railway dynamometer car, the "traction dynamometer" is used considerably in studying the design and application of steam and gas tractors.

Certain forms of dynamometers are used in production studies in factories. By using a recording instrument in connection, it is possible, in machine-shop work for example, to make complete daily records of the performance of both man and machine (see Fig. 75).

Automobile and gas-engine manufacturers are large users of dynamometers in connection with the adjusting and "running in" of new engines. If an electric dynamometer is used, any deviation from standard performance on the part of one or more motors is readily detected from the wattmeter charts.

II. THEORY

Power is defined as the rate of doing work, the horsepower being a rate of 33,000 ft-lb per min. Work is the product of two

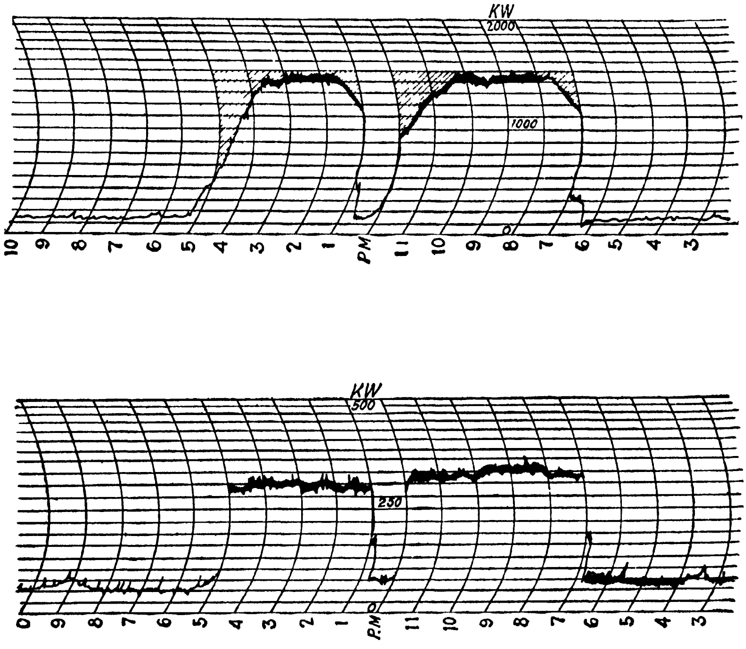


FIG. 75.—Recording wattmeter charts.

factors. In linear units the product is force times linear distance; in angular units it is torque (moment of force) times angular distance. Thus, in units of feet and minutes:

$$\text{Horsepower} = \frac{FD}{33{,}000} = \frac{T\alpha}{33{,}000} = \frac{WL2\pi\ \text{rpm}}{33{,}000}. \qquad (7)$$

Although the dynamometer is spoken of as a device for measuring **power,** it actually measures the *force* factor only, either F or W in the above formulas. This necessitates a simultaneous measurement of distance (linear or angular velocity) before horsepower can be calculated. In measuring the power being transmitted by a rotating shaft, the lever arm through which the force is acting, *i.e.*, the factor L in the last formula above, must be measured also.

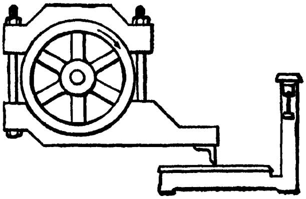


FIG. 76.—Prony brake; small wood-block type.

Absorption Dynamometers.—In general, the solid-friction brakes (Figs. 76 to 79), are used for measurements of less than 100 hp, at speeds below 1,000 rpm. These brakes may be cheaply and easily constructed by the user and are therefore favored for temporary installations. For larger powers or higher rotative speeds, a water brake or an electric dynamometer is more satisfactory. The fan dynamometer is simple and inexpensive, but its capacity is dependent upon the exact form of construction used and also to some extent upon weather conditions. Fluid-friction dynamometers and electric dynamometers have the advantage of maintaining a constant setting indefinitely, while solid-friction brakes require frequent adjustment owing to expansion of parts and to changes in the coefficient of friction. Fluid-friction brakes are in a measure self-regulating, since their torque increases with speed. They are therefore superior to solid-friction brakes for testing nongoverned engines.

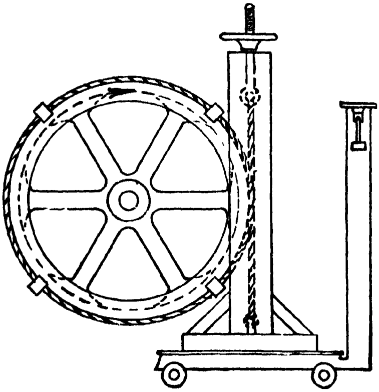


FIG. 77.—Direct-reading rope brake.

Figure 76 shows a very simple type of Prony brake consisting of two wood blocks clamped on the brake pulley by through bolts. Such a brake, if it fits the wheel properly, is satisfactory for absorbing a small amount of power. It is too clumsy to be used on wheels larger than 14 or 16 in. in diameter, and for large wheels the wood-cleat type of Fig. 78 is more satisfactory.

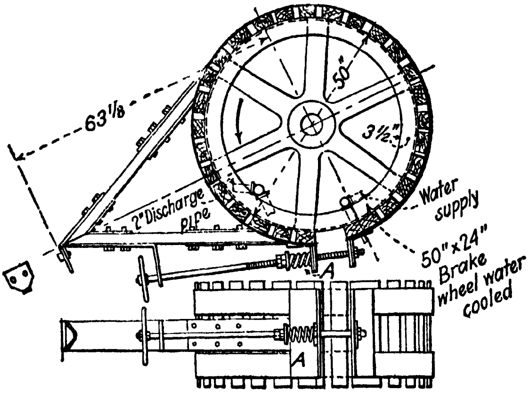


Fig. 78.—Prony brake; large wood-cleat type.

This style of brake may be used even on large flywheels and for absorbing as much as 250 hp. But the very large brakes are unusual today, as most engines developing this amount of power are used for driving electric generators, and, when a test is made, the engine generator is either tested as a unit, or the generator losses are determined separately and the generator is thus used as a dynamometer (see page 412). The combination wood block and cleat type of brake illustrated in Fig. 79 is convenient for use on brake wheels 12 to 36 in. in diameter, for absorbing less than 20 or 30 hp.

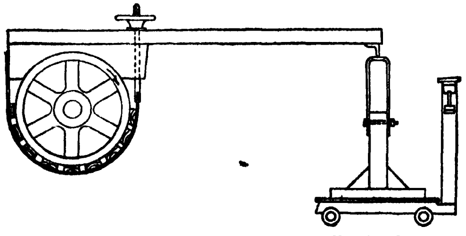


Fig. 79.—Prony brake for small wheels.

A Prony brake arm is usually made of such length that the formula $2\pi LWN/33{,}000$ reduces to WN/C, where C is some even number such as 1,000 or 500 (see theory, page 99). The length of the so-called standard brake arm, for $C = 1{,}000$, is 63.024 in., or practically 5 ft 3 in.

Brake rubbing surfaces are sometimes made of molded or of woven asbestos material such as that used for automobile brake linings. A brake lined with these materials is compact and very long lived, but it is rather expensive.

Figure 77 shows a type of rope brake which automatically subtracts the backpull or slack-side tension in the rope and therefore furnishes a single direct reading of the brake load. The tare weight of such a brake must be carefully determined by weighing the frame with the brake in place and with all ropes slack.

The **water brake** is a fluid-friction dynamometer somewhat similar in construction to a centrifugal pump. The dynamometer casing is supported on antifriction bearings and tends to revolve with the rotor. An attached brake arm, supported on scales, measures the turning moment, and the horsepower absorbed is calculated in the same way as for a Prony brake. One or more smooth rotors may be used in a water brake, or additional friction may be obtained by vanes and recesses on casing and rotor as shown in Fig. 80. The horsepower absorbed by a water brake varies approximately as the cube of the rpm and as the fifth power of the rotor diameter. A water brake is sometimes mounted on the same shaft with an electric dynamometer to produce a dynamometer unit of sufficient capacity and flexibility to test large automotive and airplane engines.

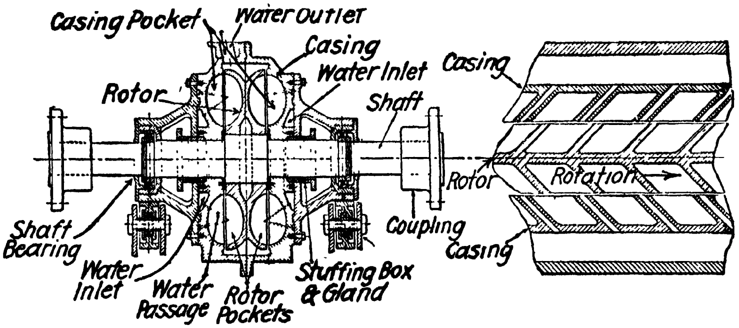


FIG. 80.—Diagrammatic section of Froude dynamometer, and developed section showing water passages.

A **chassis dynamometer** is an assembly that includes treadwheels and an absorption brake, so mounted that an automobile or a motor truck may be tested as a complete unit. The rear wheels of the automotive vehicle revolve upon the treadwheels of the dynamometer. Chassis dynamometers are made either with a ramp for floor mounting, or they may be located in a pit with the treadwheels at floor level and the controls at a convenient operating height. Several sizes are available, with capacities up to 400 hp and 100 mph.

Electric Dynamometers.—The dynamoelectric machine may be used as a dynamometer, either by cradle-mounting or by calibration. If a test or calibration is run on an electric motor, using a friction brake to absorb its output, and the results are plotted either as an input-output or an efficiency curve, these

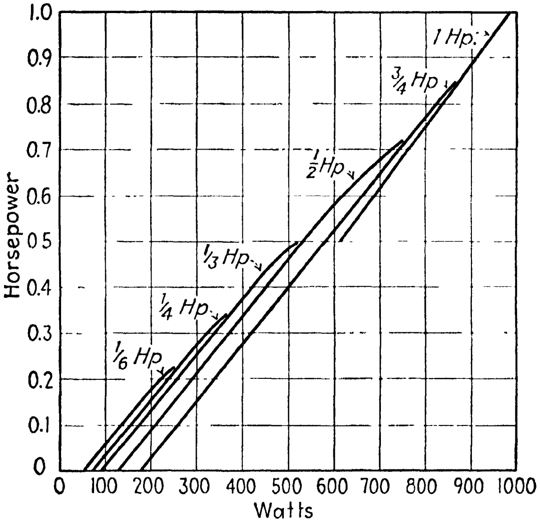


FIG. 81.—Input-output curves for typical single-phase a-c fractional-horsepower motors, 1,750 rpm.

results may be used for exact determinations of power when the motor is put into service for driving other machines. The motor then becomes a transmission dynamometer. Table IX and Fig. 81 give typical calibrations of electric motors. A generator may be similarly calibrated by the determination of losses at the various loads, and it then becomes an accurate absorption dynamometer.

For such applications as the testing of automotive engines, where a variety of speeds is necessary and it is desired to use the electric machine either as a motor or as a generator, it is much simpler to cradle-mount the machine than it is to determine and use the large number of calibration curves which would be necessary. The extended shaft of the machine is mounted at both ends in ball-bearing pedestals (see Fig. 82), and the frame or stator is left free to swing about the axis of rotation. The frame

TABLE IX.—EFFICIENCIES OF TYPICAL ELECTRIC MOTORS AND GENERATORS
(To interpolate, plot input-output curve)

Rated size, hp. or kw.	Efficiencies in per cent		
	50% load	75% load	Full-rated load
½	58	65.5	70
1	70	75	77
2	78	81.5	82.5
3	82	84	85
5	84.5	85.5	86
10	86	87	87
20	87	88	88
50	88	90	90
100	88.5	90.5	91.5
1,000	92.5	94	95
10,000	95	96	96.5

NOTE.—The above are approximate average efficiencies for plain bearing machines, moderate speed, direct current or polyphase alternating current. Ball-bearing machines or special designs may show higher efficiencies. Slow-speed machines and single-phase a-c. motors usually have lower efficiencies.

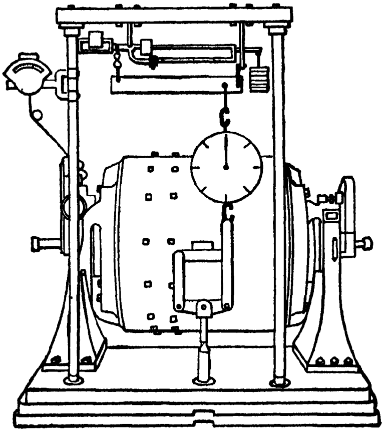


FIG. 82.—Electric cradle dynamometer. (*General Electric Company.*)

is restrained by a pair of scales or other balancing mechanism, as shown in the figure, and thus, when the machine is operated, the reaction torque (which is equal and opposite to the driving torque) may be measured. The two-way connection to the weighing scales gives a downward pull for either direction of rotation. The spring scales are interposed for convenience in load manipulation and to cushion the shocks and vibration. One of the advantages of the electric cradle dynamometer is the fact that all operations are electrically controlled, and hence the starting, stopping, and speed and load controls may be very easily and accurately handled by a single operator. Figure 83 shows the arrangement of a typical dynamometer control board. In production testing, the electric cradle dynamometer is often made recording by the application of a recording wattmeter (see Fig. 75, page 105).

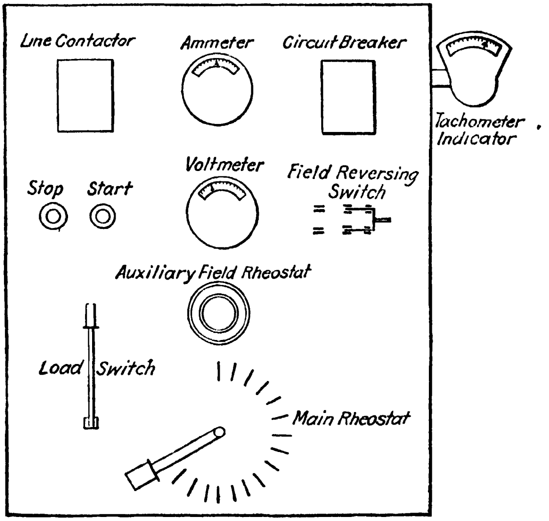


FIG. 83.—Control board for electric dynamometer.

The frame or stator of the electric cradle dynamometer is usually accurately counterbalanced so that no tare weight is indicated on the scales.

It should not be understood that an elaborate special unit is necessary when an electric dynamometer is required. Any motor, direct current or alternating current, may be cradle-mounted at small expense by fitting a longer shaft and supporting the unit with ball-bearing shaft hangers. A torque arm is attached to the frame, and the torque measured by platform scales.

OPERATING PRECAUTIONS—ELECTRIC DYNAMOMETERS

1. Before starting dynamometer, check tare weight (see page 109), and balance scales at zero. Lock stator in position during starting.
2. To start dynamometer as a motor, cut out all field esistance to give maximum field flux, and cut in all armature resistance to limit starting current. Do not push start button until field ammeter (or bull's-eye light) shows that field is on.
3. If direction of rotation is wrong, push stop button, put full resistance into field, then change field-reversing switch. Cut out field resistance before restarting.
4. To increase speed of dynamometer as a motor, first reduce armature resistance, then increase field resistance if necessary. "Normal" operation as a motor is without resistance in either circuit.
5. When using dynamometer to crank an engine, push stop button to disconnect line contactor as soon as engine begins firing. (If no stop button is provided, turn large dial switch to "off" position.)
6. If circuit breaker disconnects, stop machine and locate trouble before closing breaker.
7. To connect dynamometer as a generator (absorption dynamometer) cut in all field resistance *and* all armature resistance, then close load switch or load contactor.
8. To increase load, increase field current by reducing field resistance. Do not reduce armature resistance until necessary. Too low an armature resistance means excessive current and overheating. Watch ammeter and do not exceed current stamped on dynamometer name plate.
9. Use auxiliary field rheostat for fine load or speed adjustments.
10. Voltmeter and ammeter are for guidance in operation, and readings are not to be used for computing shaft power.
11. Always keep within speeds that are safe for both dynamometer and connected machine. In no case increase speed above that stamped on dynamometer name plate.
12. Do not leave dynamometer running without an attendant. Some accident such as a connection loosened by vibration may cause overspeeding and wreck the machine.
13. Learn to make an emergency stop, as the necessity may arise at any time. Usually this means: (*a*) If dynamometer is being used as a driving motor, push stop button or turn armature switch to "off" position. *Never pull field switch first.* (*b*) If dynamometer is absorbing power from an automotive engine, turn off ignition.

Magnetic drag or eddy current brakes are similar to cradle-mounted generators, but the electrical energy is dissipated within the machine itself. Load rheostats are therefore not necessary, and the entire installation is more compact and less expensive than that of a cradle-mounted d-c machine, but of course the motoring feature cannot be obtained in a magnetic brake. Commercial water-cooled magnetic brakes are available for automotive engine testing.

Transmission Dynamometers.—The mechanical-transmission dynamometer was developed before the advent of the electric dynamometer, and the mechanical type is now of less importance than formerly. A common form of the mechanical-transmission dynamometer is the **differential-lever type.** This machine measures, by one means or another, the difference in tension on the two sides of a belt or chain drive, or the reaction caused by the torque of a gear drive. The common characteristic of these machines is that the torque to be measured is balanced by standard weights acting with a known leverage, and for this reason this class of dynamometers is sometimes called the "weight-arm" type. The torque of the revolving shaft cannot, of course, be balanced directly, but it is balanced indirectly by measuring the reaction of the bearings that support the shaft or shafts carrying the pulleys, sprockets, or gears used in transmitting the power.

Torsion dynamometers, which measure the power being transmitted by a shaft, may utilize the elasticity of the shaft itself, or for small powers a spring coupling may be inserted in the shaft. In either case the elastic deflection is most conveniently and accurately indicated by electrical or optical methods, though mechanical or hydraulic indication is successfully used in some designs.

Several installations have been made of a torsion meter in which the angle of twist of the power-transmitting shaft is measured by the electric gage principle. This gage uses reactance coils and a bridge circuit.

Shaft dynamometers, which measure the twist or torsion of the shaft itself while it is transmitting power, find their greatest application in measuring the power being delivered by a ship-propeller shaft. While the twist of the shaft is indicated by the dynamometer, the actual torque can be determined with exactness only when a shop calibration of the shaft (before installation) is available. To make such a calibration, the shaft is supported on roller bearings, is fixed at one end, and a known torque or moment is applied at the other end or at any given distance from the fixed section, by means of lever arms and known weights or calibrated springs. The twist of the shaft over any desired section is then measured by the rotation of separately affixed pointers. By the results of this calibration the "modulus of elasticity for shear" of the test section of the shaft is determined. It will be remembered that the modulus of elasticity is defined as "unit stress over unit strain," or for torsion

$$F = \frac{s}{e} = \frac{sl}{c\theta},$$

where F = modulus of elasticity for shear (since torsion causes failure by shearing of the material), pounds per square inch.
s = shearing unit stress, pounds per square inch.
e = strain or deformation per unit of length of the shaft, inches.
l = length of the test section, inches.
c = distance from axis to the extreme or most remote fiber, inches.
θ = angle of twist of one end of the test section with respect to the other, radians.

The above relation will be understood if it is noted that $c\theta$ is the total linear deformation at the extreme section, for the entire length of shaft under consideration, and therefore $c\theta/l$ is the deformation per unit length of shaft. Equating the internal moment to the external twisting moment or torque (calling the latter M), we have

$$\frac{sI_p}{c} = M, \quad \text{or} \quad s = \frac{Mc}{I_p}.$$

Substituting this value of s in the previous equation, and noting that the polar moment of inertia I_p for a solid shaft is equal to $\pi d^4/32$, we find that

$$F = \frac{sl}{c\theta} = \frac{32}{\pi}\frac{Ml}{d^4\theta}. \qquad (8)$$

The same formula is applied when using a torsion meter on the shaft after it has been put into service, the moment or torque M being the unknown in this case. As in the case of all other dynamometers, the speed of rotation of the shaft must be observed at the same time the dynamometer readings are taken, in order that power may be calculated.

The **tractor dynamometer** is made in two forms. For the lighter types of service, such as the testing of road vehicles, steam and gas tractors, it usually consists of a calibrated spring, with an arrangement for coupling to the draw bar, and a suitable indicating and recording mechanism. For heavier service, including the determination of locomotive draw-bar pull and train resistance, a hydraulic cylinder and piston are used, and a record of the pressures produced is then made by means of an instrument similar to the engine indicator. A **railway dynamometer car** is a special test car fitted with such a tractor dynamometer and with other instruments for indicating and recording train mileage and speed, air-brake operations, etc. In some cases the test men in the dynamometer car also take complete test data on the locomotive during the test run.

EXPERIMENT 17. Inspection of Dynamometers and Determination of Constants

I. INSTRUCTIONS

1. Make a list of all the power-measuring equipment to be found in the laboratory, and name in each case the driving or driven machine and the power and speed ranges.

2. For one of the Prony brakes designated by the instructor, determine the tare weight by two methods (see Sec. II, below); determine also the length of brake arm, the brake constant, and the net scale weight for full load. Compute total area of rubbing surface and the maximum torque (lb-ft) per square foot of this area.

3. Make a sketch of the control panel for one of the electric dynamometers, indicating the function of each unit on the panel, and tracing important connections.

4. List the steps in the manipulation of this control panel: (*a*) for starting an automobile engine connected to the dynamometer; (*b*) for loading the engine (see Operating Precautions, page 106).

II. PRECAUTIONS AND INSTRUCTIONS FOR USING SOLID-FRICTION BRAKES

1. Tare Weight.—An accurate determination of the unbalanced or dead weight of the brake arm or standard is very important, as any error in this determination will affect every reading during the test. The most accurate method is to insert some form of knife-edge between the upper part of the brake and the pulley face. The knife-edge should be parallel with and directly over the axis of the shaft, the lower part of the brake being free. When this method cannot be used, the brake pulley may be turned by hand, first forward and then backward, and the weights read on the scales in each instance. The mean between the two readings is then taken as the dead or tare weight. This method is not accurate unless the brake is symmetrical. When the tare weight of a rope brake is being measured, care should be taken that tensions in the ropes do not affect the accuracy of the result.

2. Reducing Vibrations of Brake Arm.—Several methods are available, and it is often desirable to use more than one of them on a given installation. The inertia effect of a heavy weight attached to the brake arm or placed on the scales will do much to eliminate "dancing," although this also reduces sensitiveness. It is better to use a counterbalanced weight, *i.e.*, one on each side of the brake pulley.

If the adjusting nut is made to exert its pressure against a coil spring, some of the vibration will be eliminated; or a spring may be inserted between the brake arm and the scale.

The dashpot is a convenient brake regulator. This device consists of a vertical oil-filled cylinder (3 to 6 in. in diameter), in which a loose-fitting disk piston operates. The piston rod is attached to the brake arm, and the resistance offered by the device to any sudden movement may be adjusted either by varying the clearance of the piston in the cylinder, by drilling holes in the piston, or by using an oil of different viscosity in the cylinder.

The splashing of the water upon the friction surfaces, or intermittent oiling of the same should be avoided, as these practices change the coefficient of friction and make it difficult to maintain the brake load constant.

If the contact or rubbing surface is too small, the action of the brake will be irregular. On the other hand, if it is too large, the brake will be sluggish. A good rule is to allow 1 sq ft of brake surface for each 160 lb-ft of the torque applied. This is for a water-cooled brake. A noncooled brake should have an area of rubbing surface at least twice that of a water-cooled brake.

3. Cooling and Lubrication.—The heat generated by a solid-friction brake is most satisfactorily dissipated through the use of an internally flanged brake pulley carrying cooling water. One method is to provide a supply pipe that delivers cold water and a scoop with connected discharge pipe for removing the hot water from the channel of the pulley. If a large quantity of water is circulated by this method, however, the reaction of the discharge scoop will affect the accuracy of the torque readings slightly. Water may be poured into the wheel periodically and evaporation relied upon to dissipate most of the heat. If this is done, special care must be taken that the wheel never runs dry, as overheating may cause rupture of one or more spokes, due to excessive expansion of the rim. This is especially likely to occur in the case of heavy flywheel rims.

4. Brake Readings.—The proper method of taking brake readings when using platform scales is to set the scales for the desired load and then adjust the brake to maintain the load.

If the bearing point of the brake arm is not in a horizontal line with the shaft center, the tangential force will have a horizontal component and the results will not be accurate. The bearing should be between metal surfaces, well lubricated. A knife-edge parallel to the shaft, a pointed bearing pin, or a spherical surface (head of a carriage bolt, for instance) are the preferred forms of bearing, and they should rest on a metal surface on the scale platform.

It should be remembered that the power measurement involves two readings, brake load and revolutions per minute, and that these must be taken simultaneously.

CHAPTER III

LABORATORY EXERCISES

The experiments and exercises presented in this chapter are intended to acquaint the student with the various machines and types of apparatus that he will use in his later work. Such study should enable the student to fix in mind the general arrangement and proportion of parts composing the assembled device or machine. It should also furnish him, in part, a practical basis upon which to build his applied thermodynamics.

EXPERIMENT 18. Hand Hoists

Study of Mechanisms and Determination of Efficiencies

I. PREFACE

1. General.—In the order of their increasing efficiency and, incidentally also, in the order of their increasing complexity, hand-hoisting machines may be classified as follows:

Simple block and tackle.
Differential hoist.
Duplex or worm-gear hoist.
Triplex hoist or spur-geared hoist.

The principles utilized in the construction and operation of these elementary hoists furnish the underlying features and basis for the design of the more elaborate hoisting machines.

Since the hoist is designed primarily for lifting, the work expended in raising the given mass a definite distance against the force of gravity is a direct measure of the useful work performed.

2. Block and Tackle (Fig. 84).—This hoist consists of two blocks, upper and lower, which serve as frames for the mounting of two or more pulleys each. The upper or stationary block is fastened to an overhead support, while the lower, or moving block, carries the load. One end of the hoisting rope is fastened to the upper block, the other end being wound alternately over the sheave wheels or pulleys until it hangs free. It is to this end of the rope that the force *P* is applied in lifting a given load *W*.

The raising of a load is dependent upon the displacement of the lower or load block only and not upon the rotation of the sheaves on their axes. It is only from a practical consideration, that of decreasing the friction losses and wear, that the sheaves are mounted on rotating axes.

The energy losses are due to the necessity of raising the load block, hook and rope, to the friction of sheave axes and to bending of the rope.

In this type of hoist the power rope must be held to prevent reversing motion.

3. Differential Hoist (Fig. 85).—The upper block carries two sheaves cast or rigidly fastened together. These sheaves are of different diameters and have link pockets around their circumference for carrying and preventing slipping of the chain. The lower block carries but one sheave, which is grooved and not notched, thus permitting the chain to slide and therefore the block to remain at the bottom of the loop of the load chain. The chain is threaded through the blocks as in the block and tackle, excepting that the two ends are joined, making an endless combined load-and-power chain. The displacement of the lower block in this type of hoist is dependent upon the rotation of the sheaves and axes, and this positive rotation is secured because the link pockets prevent slipping.

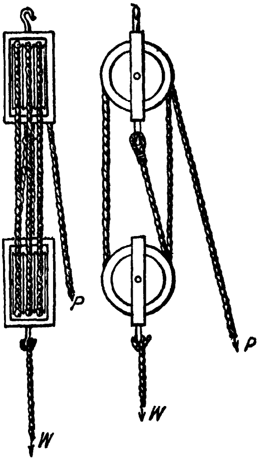


Fig. 84.— Block and tackle.

This hoist will not reverse motion of itself, the sustaining power of friction acting as a retardation both raising and lowering. This automatic sustaining power is due to the small difference in diameter of the sheaves in the upper block, the one having but one extra link pocket. This makes the multiplication ratio of power to load too small to overcome the friction of the parts and to allow the hoist to run backward or lower the load.

4. Duplex Hoist.—In this type of hoist the power is applied by an endless hand chain to a large link-pocket drive wheel which is keyed to the axis of a worm. The power is transmitted to the worm wheel and hence directly to the two-link pocket-load sheaves which are keyed to opposite ends of the worm-wheel shaft. The duplex hoist has separate hand and load chains, either of which can be lifted directly from the sheaves. The two ends of the load chain, each of which carries one-half the load, hang separately from the load sheaves and are fastened directly to the load-hook yoke. Loads are raised or lowered by application of power to opposite sides of the hand-chain loop.

The self-sustaining power of this hoist depends directly on the nonreversibility of worm and wheel.

This type of hoist as a rule requires the minimum clearance, *i.e.*, the minimum distance between hooks when the load hook is at its upper limit.

5. Triplex Hoist.—In this hoist the hand-chain power is transmitted through a direct spur-gear train, sun-and-planet motion, to the load-chain lifting sheaves.

The triplex hoist has a separate automatic sustaining device which is arranged to lock against the force of gravity and not against the applied driving force. This separation between the sustaining and applied driving force effects a saving of effort on the part of the operator in overcoming friction. This sustaining device permits, with a slight backward pull on the hand chain, a very quick action in lowering, with attendant safety. As in the duplex hoist, the hand and load chains are separate.

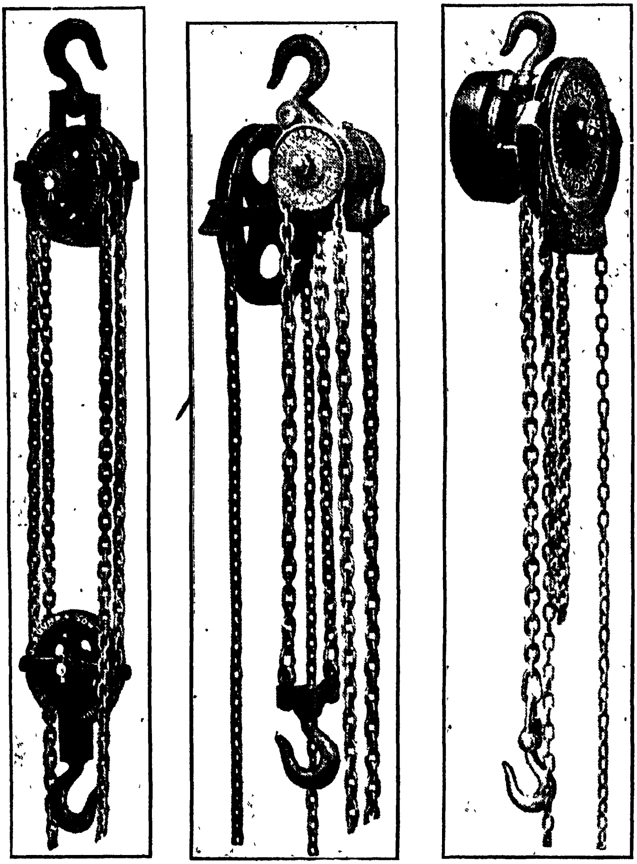


Fig. 85.—Differential, duplex, and triplex hoists.

II. ANALYSIS

In Raising:

Let w = weight on hand chain to raise weight W on load chain.
l = distance w moves through while W moves through distance L.

Ideal or theoretical mechanical advantage: $N = \frac{l}{L}$.

Actual or practical mechanical advantage: $N_1 = \frac{W}{w}$.

$$\text{Actual or direct efficiency} = E = \frac{N}{N_1} = \frac{WL}{wl}. \tag{9}$$

$$\text{The friction work} = wl - WL. \tag{10}$$

In Lowering: For reverse operation the larger force becomes the force applied and the smaller force the pull or resistance necessary to balance it. Therefore, in lowering, the load chain is the driver and the hand-chain resistance is the resistance to be overcome.

$$\text{Reversed efficiency ratio } E_1 = \frac{N}{N_1} = \frac{wl}{WL}. \tag{11}$$

When the hoist has to be assisted to lower its load then W acts in the same direction as w, and E_1 takes the negative sign, *i.e.*, the frictional resistance of the machine is greater than the useful resistance.

$$\text{The friction work} = WL + wl. \tag{12}$$

III. PREPARATIONS

1. Apparatus.—Secure 1,000 lb in standard 50-lb weights; spring-scale dynamometer of 100-lb capacity (or 100 lb of small standard weights); steel tape.

2. Setup.—Place loading platform on hook of hoist. Insert hook of spring dynamometer or hang weights in a hand-chain link at a desirable height so that the power may thus be applied to the hand chain. Oil parts, as designated by instructor.

IV. INSTRUCTIONS

1. Procedure.—After placing a known weight on the load hook, find the steady pull in pounds on the hand chain to raise the load without acceleration.

2. Determinations.—Examine and make a study of the hoists assigned. Explain with the aid of sketches the construction and operation of each hoist. (The concise description of the machines in the Preface to this experiment furnishes a general guide as to the character of presentation desired for all future experiments when a description of the operating mechanism is requested.)

Form 18.—Tests of Hoists

	No. 1	No. 2
	General description	
Type	....	. .. .
Maker	....	... .
Capacity	.. ..Lb Laboratory No	.. Lb Laboratory No.

	No. 1	No. 2
	Experimental Constants	
Range	..... Ft Clearance. . .... .Ft	......Ft Clearance........ Ft
Vel. ratio	.. ..	....
	Maximum efficiency. .. . .. %	Maximum efficiency . ... %

Experimental log

	Raising			Lowering			Raising			Lowering		
Observation, no.	Load-chain tension, lb	Hand-chain tension, lb	Direct efficiency, per cent	Load-chain tension, lb	Hand-chain tension, lb	Reversed efficiency, per cent	Load-chain tension, lb	Hand-chain tension, lb	Direct efficiency, per cent	Load-chain tension, lb	Hand-chain tension, lb	Reversed efficiency, per cent
1												
2												
3												
14												
15												
16												

Date.................... Observers:.......... ....
....
....

Determine the ideal or theoretical mechanical advantage: (*a*) by actual measurement of displacement, *i.e.*, the distance moved through by the effort to that moved through by the weight; (*b*) by calculation from the size and construction of the parts comprising the gear multiplication or leverage system.

Determine the actual or practical mechanical advantage.

Take readings for each 100-lb increment of platform load from 0 to 1,000 lb total load. Obtain constants called for on Form 18.

3. Results.—Plot direct-efficiency curve between loads raised in pounds as abscissas and efficiencies in per cent as ordinates.

Plot straight-line law curve between load raised in pounds as abscissas and effort on hand chain in pounds as ordinates.

Plot friction-variation curve between effort on hand chain to overcome friction in pounds as ordinates, to load raised in pounds as abscissas.

V. NOTES

The load and effort may be applied in a number of ways, such as, by the use of standard weights, by dynamometer or spring balances, calibrated water tank, water-tank and platform scale, etc. The most satisfactory results are obtained by the use of standard weights. The recording needle of the dynamometer fluctuates so that accurate readings are not easily obtained by its use. An objection to the other methods suggested is that the distance of movement is so small that an average value is not secured.

The distance the weight moves is very small compared with the hand-chain movement; hence the velocity ratio resulting by direct measurement may not have the desired degree of accuracy. The exact method is to count the number of teeth of the various gears and from these data determine the ratio. In the case of the differential hoist, a link pocket may be taken as the unit of measure. The velocity ratio may therefore be found by subtracting the number of pockets in the small wheel from the number in the large wheel and dividing this difference into twice the number in the large wheel.

Clearance.—By clearance is meant the distance between hooks when the load hook is at its upper extreme position.

Range.—By range is meant the total travel of the load hook between its upper and lower extreme positions.

Lubrication.—See that the hoist is oiled before making test.

EXPERIMENT 19. Pulley and Belting Study

I. PREFACE

In industrial plants a very large percentage of power is transmitted through rope and flat-belt drives of various types.

The horsepower transmitted by a belt depends upon many factors such as: kind of joint, condition, speed, width, thickness, and tension of belt; arc of contact; kind, condition of face, and diameter of pulley. Other things being equal, the belt that has the highest coefficient of friction will transmit the maximum power. The commercial belts may be listed in point of transmission efficiency as follows: leather, solid woven cotton, balata, friction surface rubber, stitched and filled canvas. The above statement does not mean that the preference should be always in the order named; the kind of service must be considered.

For outside use a rubber-coated belt is recommended. Certain manufacturers, however, claim they can produce a leather belt nonabsorptive of water and cemented with a glue inert to the action of water. Again, a balata or camel's-hair belt is to be recommended where dust or acid fumes are present.

In order to transmit power from one pulley to another by means of a flat belt, there must be a difference in the tension set up between the "tight" side and the "slack" side of the belt. The ability to set up this difference (40 to 50 lb per inch width of single belt) depends upon the coefficient of friction between belt and pulley for any given surface and arc of contact.

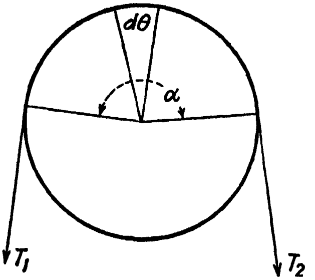


FIG. 86.

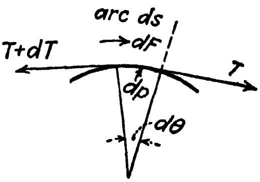


FIG. 87.

The experiment to follow is to obtain this coefficient for low tension for various kinds of belts and pulleys.

II. ANALYSIS

1. Law of Belt Friction.—In the diagrams in Figs. 86 and 87, let

T_1 = tension on tight side.
T_2 = tension on slack side.
F = total friction between pulley and belt.
θ = arc of contact in circular measure.
α = angle of arc covered by belt, degrees.

Let dF be the friction of an elementary arc ds. If dP is the normal pressure on this elementary arc, then $dF = fdP$ in which f = coefficient of friction and $T_1 - T_2 = F$.

Consider the elementary arc of the belt ds as shown in Fig. 87. The forces acting on this elementary arc are: the tensions $T + dt$ and T, the friction dF, and the normal pressure dP. Taking moments about the center of the pulley, we have $(T + dT)r = dFr + Tr$, or $dTr = dFr$. Therefore, $dT = dF$.

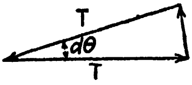


FIG. 88.

Now since dT and dF are in equilibrium on the start of motion the other remaining force T, dP and T must also be in equilibrium and will form a closed triangle of forces as shown in Fig. 88. We have approximately $dP = Td\theta$ so that $dT = dF = fdP = fTd\theta$. Integrating between limits ana simplifying

$$\int_{T_2}^{T_1} \frac{dT}{T} = \int_0^{\pi} fd\theta,$$

whence

$$\frac{T_1}{T_2} = e^{f\theta} \tag{13}$$

where $e = 2.71828+$ is base of Napierian system of logs. Now the modulus of the common system $= 0.4343$ (nearly), therefore,

$$\text{common log } \frac{T_1}{T_2} = 0.4343 f\theta,$$

since the log of a number in any system equals the modulus of that system multiplied by the Napierian log of the number.

Now since α = degrees in arc of contact and since θ = arc of contact in circular measure, we have

$$\frac{\theta}{\pi} = \frac{\alpha}{180}, \text{ or } \theta = 0.01745\alpha,$$

therefore,

$$\text{common log } \frac{T_1}{T_2} = 0.00758f.$$

In case of ropes, etc., which encircle the pulley, if n = number of turns and fractions of a turn, $n = \alpha/360$; therefore,

$$\theta = 2\pi n \text{ and common log } \frac{T_1}{T_2} = (2\pi n)0.4343f = 2.7288fn.$$

For the case when the sides of the belt are parallel, $n = \frac{1}{2}$ and we have

$$\log T_1 - \log T_2 = 1.3644f.$$

If we now plot our experimental data on log paper and take the intercept of the line on the T_1 axis where $T_2 = 1$ and $\log T_2 = 0$, we have

$$\log T_1 = 1.3644f, \text{ or } f = \frac{\log T_1}{1.3644}, \tag{14}$$

i.e., divide the log of the intercept on the T_1 axis by 1.3644 and the result is the coefficient of friction for the case considered. For further study, consult the discussion on Logarithmic Paper in Appendix.

III. PREPARATIONS

Apparatus.—The apparatus (Fig. 89) consists of three pulleys made of different materials. The pulleys are keyed to a shaft;

this shaft is pinned in the boxes of the supporting hangers so as to prevent rotation of both shaft and pulleys.

Secure one each of 10-, 20-, 30-, 40- and a number of 50-lb standard weights; also spring-scale dynamometer of 50-lb capacity.

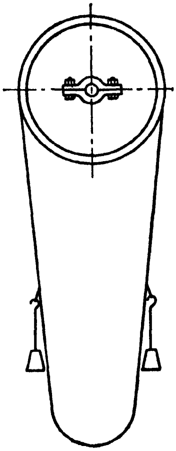


FIG. 89.—Belt friction test.

IV. INSTRUCTIONS

1. Procedure.—Various loads (standard weights) are hung from special loops attached to the belts. For each load on the slack side of belt, the load on the tight side required to move the belt downward without acceleration (after starting) is determined.

2. Precautions.—Under certain conditions of dead loading, the belt will continue to move after starting at such a slow rate that its movement will not be detected by the eye. Use a gage line at edge of belt to check movement. T_1 and T_2 equal the weights hung from their respective loops plus half the weight of the belt, including loops.

3. Determinations and Operating Conditions.—Place smooth or skin side of belts next to face of pulleys. Place standard weights from 0 to 50 lb by 10-lb increments on the slack side of belt and determine by means of the 50-lb weight and use of 50-lb spring dynamometer the load on the tight side of belt required to move it downward after starting without acceleration.

Determine weights of belts with loops.

Repeat determinations by reversing direction of motion of belts.

Reverse belt, placing rough side next to face of pulley as in English practice and determine the coefficient of friction.

Since there are three kinds of pulleys (steel, cast iron, wood) and three kinds of belts (leather, rubber-coated, cotton), there are nine distinct coefficients of friction. Consult instructor as to what coefficients are to be obtained.

Note the character of surface on belting and pulleys whether dry or oily, stiff or pliable, rough or smooth.

4. Results.—Plot curves on logarithmic cross-section paper between T_1 in pounds as ordinates and T_2 in pounds as abscissas. From the analysis as presented in the Appendix on the use of logarithmic paper, you can determine from the above plot the coefficient of friction for each combination of pulley and belt, (see Fig. 198, page 485).

5. Discussion.—Account for any difference in coefficients obtained on reversing direction of belting. Check your values for *f* against handbook values; give authorities.

Evolve analysis as to method of treatment for V-type belt, fitting grooved pulley.

V. NOTES ON BELTING

An endless belt that is cemented in place over pulleys by use of clamps will be stretched approximately $\frac{1}{10}$ in. for each foot of actual length of belt. Make this allowance when cutting belt to proper length.

Where belts are to be put on without clamps, do not deduct for stretch or the belt may be so tight it will be injured in forcing over the pulleys.

The strongest part of belt leather is about one-third the way in from the flesh or rough side. It is, therefore, desirable to run the grain or smooth side on the pulley so that the strongest part of the belt may be subject to the least wear.

The flesh side is not so liable to crack as the grain side, when the belt is old; hence it is better to crimp the grain than to stretch it.

It has been proven experimentally under normal tension conditions that when the flesh side is next to the pulley, the belt will transmit only 35 to 80 per cent as much power as when the skin side is next to the pulley.

Best practice sanctions running the inside pointed edges of lap joints toward the pulley for single belts and the reverse direction for double belts.

A properly constructed leather belt will be absolutely straight when unrolled on floor and will have all joints directly under the cross center line of the piece of leather on which the splice is made.

Laps should be of uniform thickness, or noise due to slap will result.

Belt speed should not exceed 5,000 ft per min for long life of belt.

Keep engine oils away from belts. They rot both belt and glue.

EXPERIMENT 20. Study of Oiling Devices

The importance of lubrication, the physical tests of lubricants, and the selection of and specifications for a lubricant which is to be subjected to a given set of conditions are treated in Chap. IV.

The methods of applying lubricants to rubbing surfaces, in the order of their increasing efficiency, are, in general, by intermittent application; oil cups; siphon lubricator; oil bath, and forced feed.

Intermittent feeding is accomplished by oil can or by mechanical means and is used for bearing surfaces under light pressures. Oil-can oiling is invariably wasteful and inefficient.

Oil Cups and Grease Cups (Figs. 90 and 91).—This method of oiling represents the largest number of oiling devices to be found in the small power plant. The oil cups, as a rule, are made sight feed and are regulated by means of a needle valve under hand control, giving limited feed.

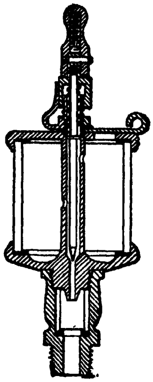


FIG. 90.—Sight-feed oil cup.

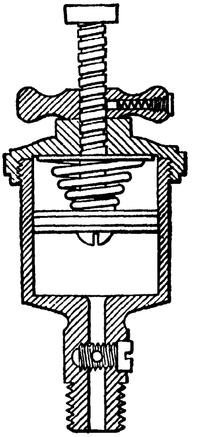


FIG. 91.—Spring-compression grease cup.

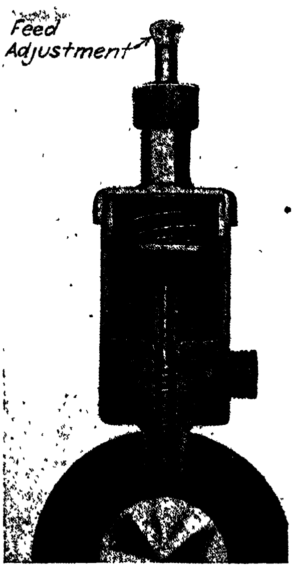


FIG. 92.—Gun-filled pressure grease cup.

With the development of pressure grease fittings and centralized systems, the soft greases are now being used for all kinds of fast-moving machinery. Figure 92 illustrates a gun-fill pressure, flow-control type of grease cup. The flow control is based on the well-known fact that more pressure is required to force the same volume of lubricant, in the same length of time, through a long pipe or passage, than through a short pipe or passage, of the same bore. The threaded worm, making a close slip fit with the sleeve, is pulled *up* into the sleeve during filling of the reservoir, thus giving a *longer thread passage* as the spring pressure increases, ensuring filling without flooding. As the spring expands during feed to bearing, the worm that moves with the piston *gradually* comes out of the sleeve, thus giving a *shorter thread passage* and reducing flow resistance to compensate for reduced spring pressures. Since the worm pitch is uniform, the variation in flow control must be uniform. The free or cross-sectional area of the passage is uniform. This design of valve gives effective control. A set of interchangeable valves is furnished so that correct feeding may be secured for both volume and type of grease.

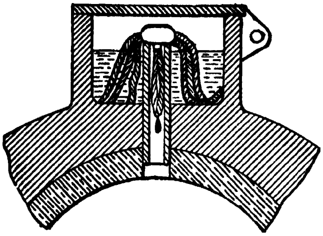


FIG. 93.—Wick or siphon oiler.

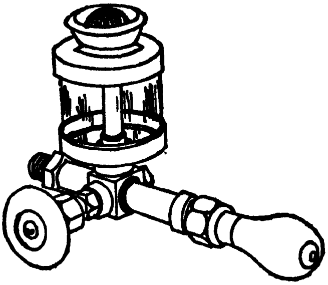


FIG. 94.—Hand oil pump.

Wick and Siphon Oiling.—Wick feed is simple yet automatic and effective. When a straight wick is used (Fig. 93), the small tubes of the wick deliver oil from the supply tank by capillarity to the rubbing surface. The supply tank may be either above or below the point of delivery. When above, capillarity is assisted by gravity, and, when below, capillarity must overcome the gravity effect. The upfeed type delivers oil only when the shaft is in operation. It is entirely automatic; the only requirement is that the wick be kept clean. In the straight type, when the receiver is above, or in the siphon type some means must be employed to cut off the flow. When the wick is bent out over the supply tank, located above the bearing surface, a siphoning effect is introduced which assists capillarity. Low-grade oils should not be used for wick feeding, as the capillary tubes will become stopped.

Forced Feed.—Lubrication by forced circulation is the best method known, where the oil is delivered into the bearings under pressure in a copious stream, collected, and returned after being filtered and cooled. Pressure and large passages ensure an ample supply, and the comparatively cool oil conveys heat from the bearing in proportion to the weight passing, the specific heat and temperature rise.

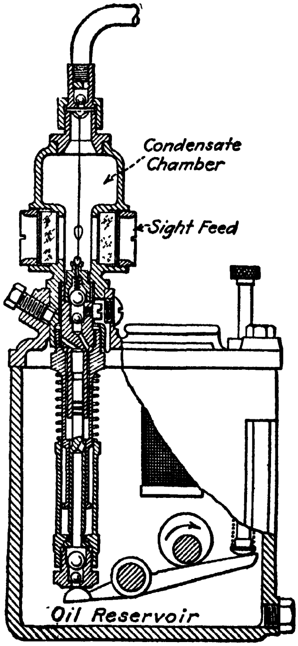


FIG. 95.—Power-operated lubricator.

In this type the oil pressure may be secured by having the supply tank above the bearings (gravity feed), or by pumping the oil through the oil-pipe system to the bearings. The pressure used varies with the style and make of oiler. The plunger-pump type is the more positive, for the pressure will build up to overcome any obstruction in the system. A pressure-feed hand pump, as in Fig. 94, is sometimes used for intermittent feed. The continuous type (Fig. 95) may be used for cylinder or bearing lubrication if it is power driven, with provision for intermittent hand operation.

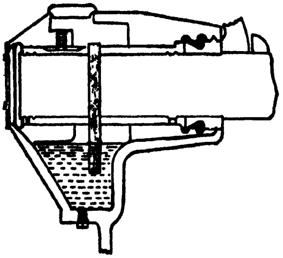


Fig. 96.—Ring-oiled bearing.

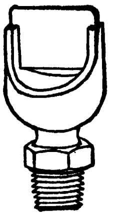


Fig. 97.—Wiper oil pipe.

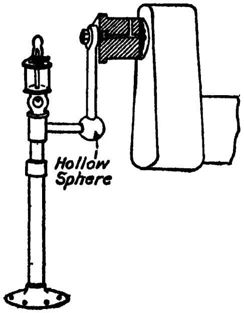
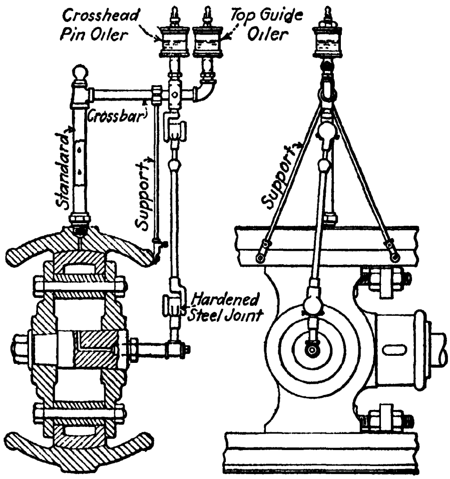


Fig. 98.—Centrifugal crankpin oiler.



Side Elevation Front Elevation

Fig. 99.—Telescopic crosshead oiler.

Durability of oil depends more on its use, or treatment, than on any inherent quality it may be supposed to possess. A rapid circulation of oil through the bearing so that it will not become overheated and vaporized is the most satisfactory and economical treatment for the oil.

Oil Bath.—The oil bath is very efficient. Other systems may become clogged, but no such failure is possible with the oil bath.

Other devices are used such as **rings** or **chains** (Fig. 96), which are fitted loosely on shaft and dip down into trough or reservoir around the bearing. As the shaft turns it carries with it by means of friction the ring or chain, and this in turn raises up a small amount of oil to the top of the bearing and from thence it is distributed. **Wipers,** as in Fig. 97, are used to lubricate crosshead pins, etc. **Centrifugal, telescopic, pendulum,** and other special types of oilers are used for crankpin oiling (see Figs. 98, 99, and 100).

Hydrostatic Lubricators.—Provision must be made for feeding oil to all inclined or horizontal steam-engine cylinders; otherwise scoring and grooving and loss of power through friction will result. Hydrostatic lubricators introduce the oil into the steam through a tube, the construction and location of which are such that the passing steam causes atomization. The effectiveness of the lubrication depends much upon the proper design and installation of the atomizing tube. Atomization assures a uniform distribution of the lubricant over the rubbing surfaces of valve, valve seat, cylinder walls, and ring grooves.

There are two types of hydrostatic lubricator, the single and the double connection (see Fig. 101). Both types depend upon the pressure of the weight of a column of water (condensed steam) on the oil surface to force the oil into and through the atomizing pipe.

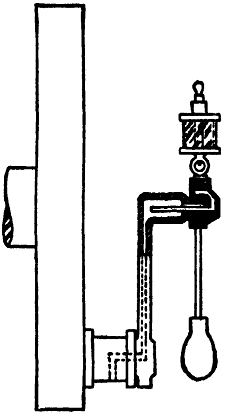


Fig. 100.—Pendulum oiler.

The longer the column of water, *i.e.*, the greater the distance of separation of the pipe connections, the more positive will be the oil feed. Two feet or more gives good results.

To Refill Lubricator.—Close water and oil-regulating valves (Fig. 101) and also top steam- and oil-discharge valves, closing the two connections to the steam line. Open the bottom drain valve to release the pressure, then remove the filling plug. Allow water to drain until the chamber is empty or until oil appears; then close drain valve. Fill lubricator and replace the filling plug.

The water valve is closed in order to conserve the head of water; otherwise this water would be drained off and the lubricator would fail to work until another head of water was condensed to supply pressure for operation.

To Start Lubricator.—Open the top steam valve. Open slowly the water valve, allowing water to flow into empty space in reservoir. If sufficient water is not present above the valve, the inrush of steam will cause the oil to foam and the sight-feed glass will then become coated. Before regulating oil flow, permit lubricator to remain idle about 5 min so that a full head of water may condense. Open oil-discharge valve and regulate oil flow by oil-regulating valve.

To discontinue use of lubricator temporarily, the oil-discharge and oil-regulating valves only need to be closed.

The single-leg lubricator may be operated the same as above explained, but the top steam valve will be absent.

Notes.—Do not permit lubricator to empty of oil completely. The sight-feed glass will then become coated with the final particles of oil and render cleaning necessary.

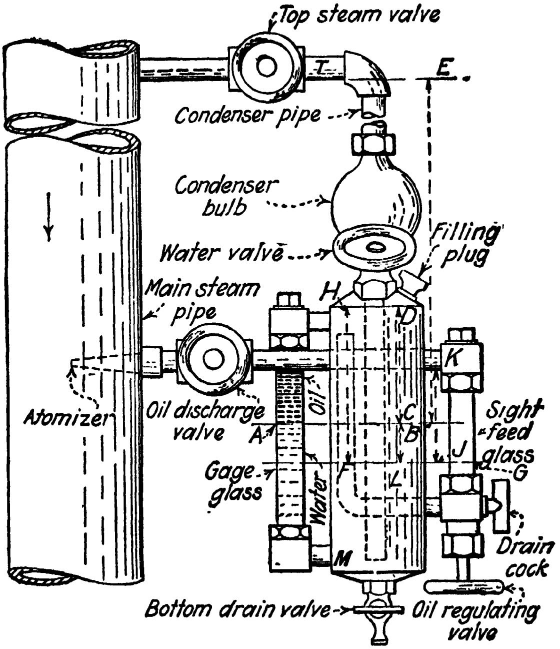


Fig. 101.—Hydrostatic steam-engine lubricator (double connection.)

If cleaning of sight glass is necessary, it may be blown out by shutting oil-regulating valve and opening sight-feed drain cock. Otherwise, after shutting the oil discharge and regulating valves, remove plug *K* (Fig. 101) and clean by use of rag on small stick. After cleaning, fill glass with water, screw in plug, and adjust as previously explained.

If bubbles of oil are too large, add a little salt or soap to water in sight-feed gage glass. See also that oil feed cone is clean and polished.

The sight glass at all times serves as an indicator to the operator as to the quantity of oil in the reservoir.

Assignment.—Examine and describe the devices used to oil the crosshead and crankpins of three of the laboratory engines which use different methods of oil delivery. Examine, fill, and put into operation the lubricators assigned for study. Regulate closely for specific number of drops per minute. Follow detailed instruction, as given above. Make a classified list of all the lubricators and lubricating systems found on machines in the laboratory.

EXPERIMENT 21. Examination of Speed-reduction Unit

Form 21.—Report on Speed-reduction Units

Gear-reduction Unit

Manufacturer..............................Size.................
Lubrication...
Single reduction......Double reduction......Triple reduction........
Parallel shafts...................Right angle shafts.... ....
Case (housing): Open....Enclosed....Material....Size of bed........
Height of high-speed shaft........Height of low-speed shaft..........
Gears: High-speed pinion:
  Kind: Spur........Helical (single or double)......................
    Herringbone (continuous or staggered)..........................
    Bevel (plain or spiral)...............Worm..................
  Material...
  Ratio, gear to pinion......Center distance, gear to pinion..........
  Number of teeth.................Diametral pitch...............
  Teeth, cut or cast...............Tooth form...................
  Provision for thrust..
Intermediate gear:
  Same items as for pinion.
Intermediate pinion:
  Same items as for pinion.
Low-speed gear:
  Same items as for pinion.
Shafts: High-speed shaft:
  Size..........................Length, protruding end..........
  Bearings, type...................Provision for coupling...........
Intermediate shaft:
  Size..........................Bearings, type.................
Low-speed shaft:
  Same items as for high-speed shaft.
Efficiency of gear unit by test.................................

Multiple V-Belt Reduction Unit

Manufacturer............................Size................
Diameter of high-speed sheave.......Diameter of low-speed sheave..
Number of belts........Size........Angle of grooves............
Center distance..........Arc of contact on high-speed sheave......
Arc of contact on low-speed sheave......Efficiency of drive........

The normal speeds of steam turbines and electric motors are in many cases too high for direct driving of industrial machinery. Some type of reducing mechanism is required. The recent adoption of multiple V-belts and various types of gear reductions has served to solve the problem in a very satisfactory manner.

Industry has made numerous applications of these reducing devices. The use of these dependable units makes possible much simpler power assemblies through the use of the smaller high-speed driving units.

Spur, herringbone, spiral, bevel, and worm gears are incorporated in the various gear-reduction units. Reductions from 2 to 1 to 80 or more to 1 may be readily purchased. A recent development is the housing of an electric motor and a reducing gear in the same case.

Assignment.—Examine and report on speed-reduction units as indicated by instructor, using Form 21 on page 127 for results.

To test the efficiency of the unit, a very satisfactory method is to direct-connect a calibrated electric motor to the high-speed shaft and measure the output from the low-speed shaft by means of a Prony brake. Load variations are readily secured by this method. A cradle-mounted driving motor (electric dynamometer) is of course even better.

EXPERIMENT 22. Steam-engine Examination

I. INSTRUCTIONS

Study Form 22 carefully and make out a list of instruments and tools necessary to secure the required information. Obtain these tools at the instrument room. Make an inspection of the engine assigned by instructor, using Form 22 for report.

Report also on action of governor in regulating speed of engine, on system of lubrication, and on method of taking up wear.

Do not remove any parts from engine unless detailed to do so. The purpose of this exercise is to establish the general size and proportion of parts for a unit of the class considered.

II. NOTES

The flywheel rim stress is (formula as used by Fidelity and Casualty Insurance Company)

$$F = \frac{WV^2}{2.66} \qquad (15)$$

where V = velocity of rim, feet per second.
W = weight in pounds of 1 cu. in. of metal.
F = force in pounds per square inch tending to burst wheel.

The speed at which a given material will disrupt (regardless of the amount of material) is expressed by the formula (Fidelity and Casualty Company)

$$V' = 1.63\sqrt{\frac{S \times E}{W}} \qquad (16)$$

Form 22. Steam-engine Report

Name of engine..................
Designed initial pressure...lb. gage
Usual r.p.m. of engine.............
Horsepower rating................
Diameter steam supply pipe.....in.
Diameter exhaust pipe..........in.
Ratio crank to connecting rod......
Diameter of crank shaft in bearing, ...in.
Diameter of crank shaft outside bearing.....................in.
Length of crank shaft bearing...in.
Type of governor used............
Number valves effecting steam distribution......................

Manufactured by................
Diameter cylinder.............in.
Length of stroke...............ft.
Diameter piston rod...........in.
Length connecting rod.........in.
Width face flywheel.........in.
Width belt pulley.............in.
Thickness of belt..............in.
Width of belt.................in.
Diameter of flywheel..........in.
Diameter of belt pulley....... in.
Velocity of belt.............f.p.s.

The flywheel rim stress F......pounds per square inch
Bursting speed V'............................f.p.s.
Flywheel speed factor $\frac{V'}{V}$....................
Flywheel safety factor $\frac{S}{F}$.....................
Engine speed at bursting.....r.p.m.

The engine is
Horizontal......vertical.......
Condensing......non-condensing........
Compound......simple........
Automatic cut-off......throttling........
Running over......running under........
Self-contained......not self-contained........

Date................ Observers:...............

....

....

where V' = bursting speed, feet per second.
W = as defined above.
S = tensile strength of material, pounds per square inch.
= 10,000 lb per sq in. for cast iron in large castings (dependable value), if test bars give an ultimate strength of 20,000 lb.
E = efficiency of rim joint.
= 1 for solid rims.

EXPERIMENT 23. Placing Engine on Dead-center Positions

The steam engine has two dead centers, the head end or inner dead center, and the crank end or outer dead center.

Approximate Setting.—For a certain class of work the placing of the engine on its approximate dead-center position gives a sufficiently accurate setting. This may be done by turning over the engine in the direction in which it runs until the motion of the crosshead stops or first comes to the visible end of its stroke, or by another method until the connecting rod and crank fall on the same horizontal line.

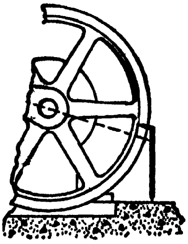


FIG. 102.—Marking flywheel for location of engine dead centers.

It will be noted after the engine is placed on a dead center by either of the above methods that the flywheel may be rotated through a considerable arc before the crosshead shows any visible displacement. The valve motion (controlled by the eccentric) is close to its maximum velocity of travel at the dead-center positions and, while the crosshead may show no apparent displacement, the valve may have received an appreciable displacement.

For the above reason it is necessary for accurate valve setting that the engine should be placed on its exact, geometrical dead center.

Exact Setting.—Turn the engine over until the piston is near the end of its stroke, make a vertical continuous pencil mark across both the crosshead and one of the guides. On the bed plate, frame of engine, or on any other stationary part, locate a prick punch mark such that if one leg of a trammel (Fig. 102) or pair of dividers be set in this point, the other end may be extended to reach the main-shaft flange coupling or the flywheel. With the free end of the dividers scribe a line on the flange or face of flywheel. (Use chalk and pencil in making scribe line.) Then turn the engine ahead of its dead center in the direction the engine runs until the scratch mark on the crosshead and guide lines up again. With the dividers or trammel, with its points at the same distance apart as before, place one end on the stationary point and scribe a new line on the rotating member. Now, if the engine is turned back so that the end of the dividers or trammel is at a point halfway between the two marks made on the rotating member, the engine will be set accurately on the center required.

Report on method of setting a vertical engine on exact dead centers.

NOTES

It is advisable first to go over the engine carefully and eliminate all possible lost motion by making proper adjustments.

Note that turning the flywheel in the direction the engine runs takes up wear in the opposite direction to that which is the case when the engine is moved under its own power. Interpret and apply this fact to your problem.

EXPERIMENT 24. Determination of Steam-engine Clearances

INSTRUCTIONS

Determination of Linear Clearance.—Place the engine on the dead center for which the clearance is desired, using the angular method described on page 130. Disconnect the connecting rod and measure in inches the linear distance that the crosshead must be moved from its dead-center position before bumping occurs. This is termed the linear or "bumping clearance."

Determination of Volumetric Clearance.—Volumetric clearance is the volume in cubic inches contained between the face of the piston and the head of the cylinder when the piston is at the end of its stroke, plus the volume of the port passages up to the valve seat, plus the volume of the indicator connections up to the shutoff cocks. It may also be expressed in percentage of piston displacement instead of in cubic inches.

To determine volumetric clearance, first place the engine on one of its dead centers by the angular method. Disconnect the valve from the valve rod, and block the former in place so as to cover the steam ports symmetrically. A thin gasket may be inserted between the valve face and its seat, thus closing off both ports and eliminating the possibility of leakage to steam chest or exhaust.

Fill the space between the piston and the end of the cylinder with water, through the indicator connections. The water is poured in until it rises to the level of the indicator cock, care being taken to avoid trapping air. Permit any leakage by the piston to drain through an open pet cock on the opposite end of the cylinder, back into the container supplying the water. Observe the weight of water required to fill the clearance space, and the water temperature, and convert from weight to volume, using the water density from Table XLIII, page 473. Calculate percentage clearance as the ratio of clearance volume to piston-displacement volume.

If the leakage is troublesome and a close determination is desired, remove the cylinder head and putty all leakage openings; or determine the rate of leakage when full and correct the original determination, assuming the same leakage rate. A leaky piston at the dead-center position may be sufficiently tight for a satisfactory determination if advanced a fixed amount, say 1 in. The true clearance could then be computed.

Volumetric clearance in percentage of piston displacement varies approximately as follows for the various types of engines: Piston valves: in head of cylinder 3 to 8; at side of cylinder 7 to 15. Flat slide valves: in head of cylinder 4 to 8; at side of cylinder 5 to 15. Corliss valves: 2 to 8. Clearance in high-speed engines is usually twice that in slow-speed engines. Linear clearance for the piston varies from $\frac{1}{8}$ to $\frac{3}{8}$ in. to allow for roughness of the casting at each end of the cylinder and for the working joints between piston and shaft.

EXPERIMENT 25. Steam-engine Valve Study—Simple D-slide Valve

I. PREFACE

The simplest steam valve is known as the "D-slide type." Figure 103 shows a D-valve in its mid-position. It will be seen that as the upper casting or slide part of the valve moves from left to right and then reverses its stroke, it will alternately admit steam through the steam passages A and B around the ends of the slides and into the ends of the cylinder and also open these ends at the proper time to the common exhaust.

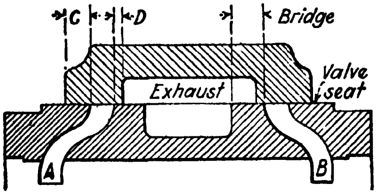


Fig. 103.—D-slide valve.

Steam Lap.—When the valve is in its center position, the linear distance C which the valve must be moved to admit steam is called the steam lap.

Exhaust Lap.—When the valve is in its center position, the linear distance D which the valve must be moved to open the exhaust is called the exhaust lap.

Steam lead is the linear amount the steam port A or B (Fig. 103) is open when piston is at the end of its stroke (see Fig. 104).

Exhaust lead is the linear amount the exhaust port is open when piston is at the end of its stroke (see Fig. 109).

Angle of advance may be defined as the angle swept through by the center line of the eccentric while the valve moves from its mid-position a linear distance equal to the lap plus the lead. It is equal to the angle of lap plus the angle of lead. **Throw of eccentric** is equal to the eccentricity, *i.e.*, the distance between the center of the crankshaft and the center of the eccentric. **Valve travel** is equal to twice the throw of the eccentric.

Port opening equals the amount the steam port is opened when the valve is at the extreme end of its travel.

Face of valve is equal to the steam lap plus the width of steam port plus the exhaust lap.

Proper distribution of the steam demands that the eccentric arm be always ahead of the crank by at least a 90-deg angle for outside valves

Analysis of Valve Operation for a Complete Forward Stroke of Engine.—Figure 104 shows the starting position when the crank is at its inner dead point. The inlet port is slightly open (the amount of the steam lead) and the crank-end port open to exhaust. The position *b* of eccentric is seen to be 90 deg plus angle *eob* (angle of lead) ahead of crank. Figure 105 gives the position of crank and valve arms when the steam port is full open and the exhaust port on opposite end full open, the valve being at the end of its stroke. Figure 106 shows the valve on its return stroke. It has just cut off the steam port on the head end. The steam in the opposite end of cylinder is being forced out of the exhaust owing to its own pressure and that due to the advancing piston. Figure 107 gives the relative positions of moving parts when the exhaust on the crank end has just been closed The

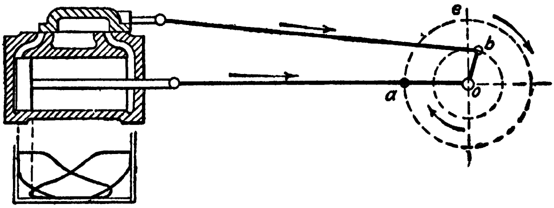


Fig. 104.

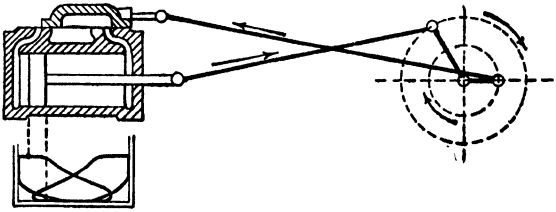


Fig. 105.

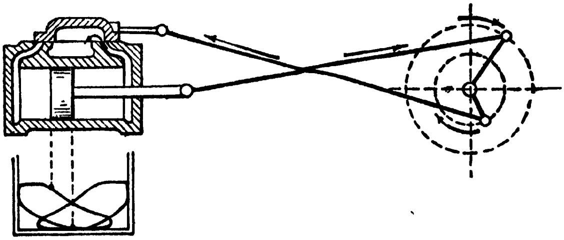


Fig. 106.

Figs. 104–109.—Analysis of valve operation.

piston has almost reached the end of its stroke in Fig. 108. The valve has moved ahead to the point of exhaust opening for the head end. Figure 109 shows the head end exhausting and the crank end on the verge of admitting steam, the piston having reached the crank-end dead-center position.

The indicator diagrams directly under the figures show the relative positions of piston in the stroke, locating the events in the cycle.

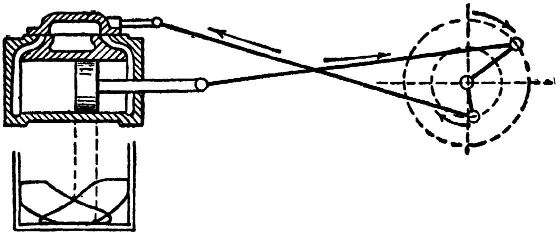


Fig. 107.

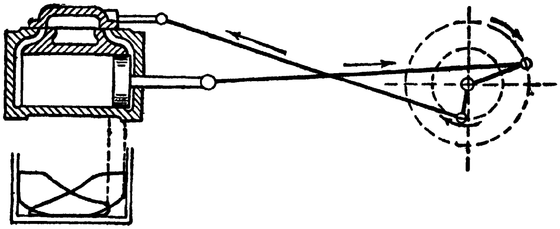


Fig. 108.

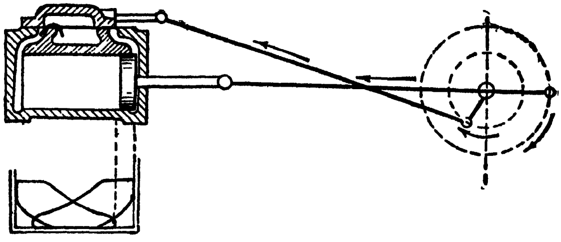


Fig. 109.

Figs. 104–109.—Analysis of valve operation. (*Continued.*)

The back stroke may be similarly treated as above.

It is evident that a valve with no steam or exhaust laps will take steam full stroke and develop an indicator card which is rectangular in shape with rounded corners produced from throttling of both inlet and exhaust ports (termed "wire drawing") (see Fig. 71, page 97). Valves of this type are used in the construction of some steam pumps and are very uneconomical, as the expansive force of the steam is not utilized in the development of power. To secure the expansive force of steam, the inlet port must be closed before the end of the stroke is completed. This is obtained in the construction of the valve by extending the edges over the steam ports when the valve is in its mid-position.

In order to bring the moving reciprocating parts to rest without jar, a cushion of steam must be supplied behind the piston. To secure this steam cushion, the exhaust valve should close before the end of its stroke, thereby trapping some steam for compressing by the returning piston. The above is accomplished by extending the lug on the exhaust edge of valve.

If the valve did not admit steam until just as the piston started back, steam could not be admitted through the small opening of the port fast enough to have full steam pressure in the cylinder at the beginning of the stroke. It is therefore customary to have the valve uncover the steam port just before the beginning of the stroke, thus causing preadmission and giving "lead."

II. INSTRUCTIONS

Read Form 25 and determine what instruments and tools will be required in securing the information requested. Obtain these at the instrument room.

Remove steam chest cover and make study of the valve motion. (Consult instructor as to method to be used in removing valve.)

Determine method used in decreasing effective area on back of valve subjected to the main-line pressure which forces face of valve against its seat. What ratio is this to the full projected valve area?

III. NOTES

To ascertain the outside or steam lap, subtract the measurement in inches between the extreme outside edges of the head and crank-end steam ports of the valve seat from the over-all length of valve and divide this difference by 2. To ascertain the inside or exhaust lap, from the measurement in inches between the inner edges of the head and crank-end steam ports of the valve seat, subtract the straight-line distance between the inner edges of the exhaust arch of the valve and divide this difference by 2.

To determine the phase angles (total angle of admission, etc.), turn the engine over in the same direction in which it moves when running, until the admission port is just opening. Square up from the engine base, floor, or some stationary level surface, through the center of the crankshaft and scribe a pencil line on edge of shaft or flywheel rim. Continue turning the engine over in the same direction until the admission steam port closes or cutoff takes place. Square up as before and scribe another pencil line. Obtain the angle between these two lines by use of a tape, measuring distance between the two scribe lines, and find the ratio of this distance to the total circumference of shaft or flywheel. This ratio, multiplied by 360 deg, will give the total angle of admission. Determine the angle of expansion, release, and compression in like manner as above.

To determine the "angular values" of the events of the cycle, consider either of the dead-end points as the starting or zero-angle position (movement to be made always in the direction the engine runs).

Use chalk and pencil, only, in making all scribe lines.

Form 25.—Steam-engine Valve Report

Simple Slide Valve—Fixed Eccentric

Type of valve gear.............................Kind of valve.......

Travel of valve........................in. Eccentricity...... in.

Diameter piston valves................in. Length. ....in.

Over-all dimensions D-valve: Length.....in. Width........... in.

Steam-port dimensions: Width...........in. Length...........in.

Exhaust-port dimensions: Width.........in. Length...........in.

Steam lap.............................in. Exhaust lap.......in.

Cut-off, percentage of stroke..

Cut-off angular value.......................................deg.

Compression, percentage of stroke....................................

Compression angular value...................................deg.

Exhaust-release percentage of stroke.................................

Exhaust-release angular value...............................deg.

Admission, percentage of stroke......................................

Admission angular value.....................................deg.

Maximum steam-port opening area...........................sq. in.

Maximum exhaust-port opening area.........................sq. in.

Total angle of admission....................................deg.

Total angle of expansion....................................deg.

Total angle of release......................................deg.

Total angle of compression..................................deg.

Date................ Observers:.............

....

....

EXPERIMENT 26. Setting the D-slide Valve—Steam Chest Cover Removed

I. PREFACE

Steam distribution may be affected by a change of any of the following valve adjustments:

1. Eccentricity.
2. Angle of advance.
3. Length of valve stem.
4. Steam and exhaust lap.

In the common D-slide valve, items 2 and 3 are usually adjustable, but, in some cases, 2 is also fixed, as must necessarily be true always for 1 and 4. Shifting valve on stem or changing length of valve stem evens up the events on the ends of the cylinder, *i.e.*, distributes the steam evenly between head and crank ends of cylinder. Changing the angle of advance changes the time of the events, *i.e.*, makes early or late cutoff, release, etc.

For plain slide-valve engines the cutoff is fixed between ½ and ⅞ stroke. Good practice is to compress to about $\frac{8}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ for medium-speed and $\frac{2}{10}$ to $\frac{3}{10}$ for slow-speed engines.

The D-slide valve may be set for either equal lead or equal cutoff, but not for both, on account of the angularity of the connecting rod. In throttling engines the equal-leads setting is customary.

To Set the Valve for Equal Leads (Set-screwed Eccentric Engine Running Over. *First Operation.*—The valve should be placed on the valve rod so that the distance of its travel will be the same on both sides of its mid-position. To so set the valve, turn the engine, or better, move the eccentric on the shaft until the valve is at one extreme end of its throw. Next, adjust the length of the valve stem to give maximum port opening. Shift the eccentric to the opposite limit of its travel and measure the width of the uncovered steam port at that end. If the widths measured at the two ends are not the same, the valve must be moved on its stem a distance equal to half the difference between the widths of the uncovered ports. The distance that the valve must be moved for equalization will be in the direction toward the port having the larger opening. Check again for equal port openings and repeat above procedure if greater accuracy of setting is desired.

Second Operation.—Put the engine accurately at the dead center of either crank or head end (see Experiment 23, page 130).

Third Operation.—Move the eccentric on the shaft in the direction in which the engine is to run until the steam port has been opened a distance equal to the lead required. In this position the eccentric should be firmly fixed to the shaft.

Fourth Operation.—Turn the engine over to the other dead center and measure the lead at that end. If the leads are not the same, the difference should be halved, one part to be taken up by moving the valve on its stem and the other by moving the eccentric on the shaft.

To Set the Valve for Equal Cutoffs.—In general, the valve should be set for equal leads, but sometimes an equal cutoff setting may be desired.

First Operation.—The valve is first set on its stem so that the travel will be the same on both sides of its mid-position. To so set the valve, turn the engine, or better, move the eccentric on the shaft, until the valve is at one extreme end of its throw. Mark the position of edge of valve and measure its distance from the center line representing the mid-position of valve when centrally located. Turn the eccentric over to the opposite extreme position of its throw, marking and measuring the distance of the other edge of valve from mid-position. If these measured distances are equal, the valve is set for equal travel. If these measurements for the two ends are not the same, the valve must be moved on its stem a distance equal to half the difference between the two measurements. The distance that the valve must be moved for equalization will be in the direction toward the side having the shorter measurement.

Second Operation.—To obtain the same cutoff at both ends, mark on the crosshead accurately the limits of the stroke and set the crosshead at the percentage of the stroke desired for cutoff; *i.e.*, if the stroke is 12 in. and the cutoff is to be 70 per cent, put the crosshead 8.4 in. from the beginning of the stroke.

Third Operation.—Move the eccentric on the shaft in the direction in which the engine is to run, if the valve takes steam on the outside (and in the opposite direction if the valve takes steam in the inside) until the valve is just closing the steam port at the end of the cylinder from which the piston is moving. In this position the eccentric should be firmly fixed to the shaft.

Fourth Operation.—Turn the engine over and observe if the valve is just closing the other steam port when the piston has moved the same distance from the other end of the cylinder. If the port is not just closing, the error should be halved, correcting for one-half by moving the valve on the stem and for the other by moving the eccentric on the shaft.

It will be found that with equal leads the cutoff will be later on the out than on the instroke, or, with equal cutoffs, the lead and port opening head end will become too small and the lead on the crank end too large. The same trouble will be found in attempting to balance the exhaust sides of the valve. The thing to do is not to attempt to exactly balance any one item in the two ends but to get the best combination of leads, cutoffs, etc., as shown by the indicator card.

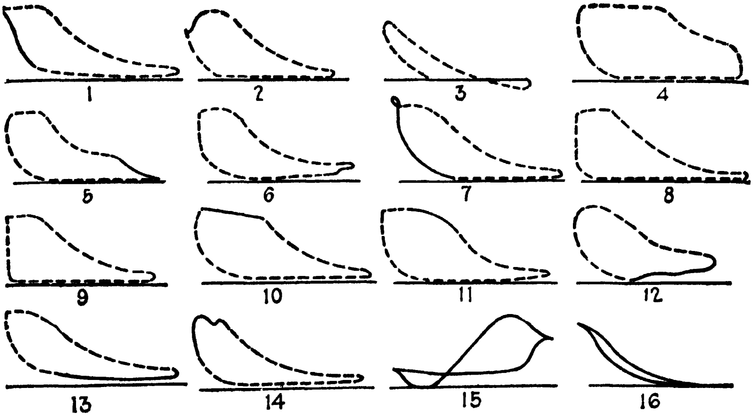
After the valves are set, indicator cards should be taken from both ends. By observing points of admission, cutoff, release, and compression, and the shape of the card, the correctness of the valve setting can be checked. The cards will also show the distribution of work between the two ends of the cylinder.

After determining the correct setting, shift the eccentric by 30-deg increments and take indicator cards at each position at which the engine will run, covering the full 360 deg. Locate events of stroke on each card, using a uniform system of marking. Make note on the cards of the angle of advance and direction of motion of flywheel.

Construct diagram of crank and eccentric arms, showing approximate eccentric angles where the engine will run over or run under. (Do not try to get the limiting position of these angles.)

II. NOTES

As previously stated, the indicator diagram may be used to determine the valve action. The accompanying diagrams (Fig. 110) are characteristic of the various derangements of valve adjustments. Distorted diagrams may also be produced by other erratic conditions such as faulty valve action, condensation, water in indicator connections, sag in indicator cord, indicator striking stops, too weak drum spring, knot in indicator cord running around drum or reducing motion, faulty reducing motion, indicator cord not running parallel to piston rod thereby changing its angularity and hence its instantaneous speed for different points in the stroke for certain types of hookups, etc.



1-Too early admission
2-Too late admission
3-Too early cut-off
4-Too late cut-off
5-Too early release
6 Too late release
7-Too early compression
8-Too little compression
9-No compression
10-Choked admission
11-Leaky cut-off
12-Choked exhaust
13-Excessive back pressure
14-Double admission
15-Eccentric slipped backward
16-Eccentric too far ahead

FIG. 110.—Errors in indicator cards.

Unequal steam distribution as well as lack of symmetry in valve setting for the two ends of the cylinder may be discovered by a comparison of the two cards.

Check up the following statements as to their correctness:

1. In an engine having a valve with no laps, no leads or angle of advance, the valve must be in its mid-position and just covering the steam ports when the piston is at the end of its stroke. In this position the eccentric arm must be ahead of the crank by a 90-deg angle in the direction of rotation.

2. If the engine has steam lap and lead, then the angle between the eccentric arm and the crank will be 90 deg plus the angle of advance.

3. To reverse the direction of rotation of the engine the eccentric must be turned backward 180 deg plus two times the angle of advance. This will bring the eccentric ahead of the crank when the engine turns in the opposite direction.

4. All the above applies to outside valves (D-slide or piston). If the valve is of the inside type, then the center line of the eccentric will be behind or follow the crank, instead of being ahead, an angle of 180 deg.

5. The ordinary slide valve cannot be set to give both equal cutoffs and equal leads because of the angularity of the connecting rod; for a valve with equal laps this angularity will cause cutoff to come earlier on the crank than on the head end

6. If cutoff occurs earlier than mid-stroke, compression will be excessive, if later than ⅞ stroke, then there will be too little expansion. Hence, point of cutoff for plain slide-valve engines are fixed between ½ and ⅞ stroke.

7. The angular measures of the expansion and compression phases are equal.

8. Too little lead may cause shock and hammer.

9. All things considered, the least lead that will produce quiet, smooth operation is the most economical.

EXPERIMENT 27. Setting Steam-pump Valves

INSTRUCTIONS

Simple, Direct-acting, Nonexpansion Steam Pumps.—In some direct-acting pumps the valves are designed without any valve adjustments, while in others the adjustments are made on the outside valve gear in various simple ways.

Duplex, Direct-acting Steam Pumps.—*a.* When the steam valve is moved by a single valve-rod nut: Place one piston in the middle of its stroke and disconnect the link from the head end of the valve rod on the opposite side. Then set the valve in its central position. Place the valve nut evenly spaced between the lugs on the back of the valve, screw the valve rod in or out until the eye of the valve-rod head comes in line with the eye of the valve-rod link; then reconnect. Repeat the operation on the opposite side, and the valves will be properly set.

b. When the valve has more than one lock nut: Place one piston in the middle of its stroke and the opposite slide valve in the central position. Adjust the lock nuts so as to allow about $^3\!/_{16}$-in. lost motion on each side of the lug, and the valve is set. Do not disconnect the valve motion. Repeat the operation on the other side.

The best way to divide the lost motion equally is to move the valve each way until it strikes the nut or nuts, and see if the port openings are equal. Too much lost motion will tend to lengthen the stroke and may cause the piston to strike the cylinder heads, while with not enough lost motion the stroke will be perceptibly shortened.

It is advisable to place both pistons at the middle of their strokes before touching either slide valve. To set the piston in the middle of its stroke, open the drip cocks and move the piston by prying on the crosshead until it comes in contact with the cylinder head. Make a mark on the piston rod at the face of the steam-end stuffing-box follower. Move the piston back to contact at the opposite end and make a second mark on the piston rod halfway between the first mark and the follower. If the piston is moved until the second mark coincides with the face of the same follower, it will be at the middle of its stroke.

EXPERIMENT 28. Setting Corliss-engine Valves

INSTRUCTIONS

Setting for Single- or Double-eccentric, Releasing Gear.—Note whether the wrist plate moves through equal angles from its mid-position. If not, equalize by adjusting the length of the eccentric rod. Set the wrist plate in its mid-position. This should also be the central position of the eccentric. A center line through the rocker arm should be exactly vertical as tested by a plumb bob. Scribe a line on the wrist-plate hub parallel to the axis of the stud, and extend it on the stud. Then set the eccentric and the wrist plate to each of its extreme positions as determined by the throw of the eccentric, and extend on the stud the same original scribe line that was made on the wrist-plate hub. The eccentric rod should now be adjusted until these two extreme travels of wrist plate from central position are equal (or the distance between scribe lines is equal). Hook up the steam valves for mid-position of the wrist plate and adjust the length of the valve rods to secure the predetermined amount of steam lap for both ends. These laps are usually marked on the valve and valve seats by the manufacturers. Do likewise for the exhaust-valve laps. The steam-valve laps for small engines vary from $\frac{1}{16}$ to $\frac{1}{4}$ in. and for larger engines from $\frac{1}{4}$ to $\frac{9}{16}$ in.

Next place the engine in turn on its inner and outer dead centers, and, with the reach rod hooked up, adjust the eccentric for the desired lead, equalizing for the two ends by changing the steam-valve link. This latter change will change the lap slightly, but its effect will be negligible in the operation of the engine.

Adjustment of Dashpot Rod.—Remove the safety plug from the governor and rock the wrist plate until both dashpot rods are released. Adjust the length of the dashpot rods until the edge of the steam valves comes opposite the steam-lap mark. These rods must have the proper lengths, for if too short the valves will fail to open and if too long the rods will jam and bend.

Adjustment of Governor Rod.—Place the engine on outer or head-end dead center, block the governor in its highest position, and unhook the wrist-plate rod. Adjust the governor cam rods, if necessary, so as to just prevent the latch from engaging when the wrist plate is at its extreme throw position. Hook up the wrist plate and set the governor at its lowest level. Place the engine on the predetermined head-end cutoff position. Raise and block the governor at the position where the latch is just released. Then place the engine at the corresponding crank-end cutoff position, and the latch should just release, permitting the steam valve to close. Adjustment may be made if necessary by changing the length of the governor cam rod.

Start the engine and verify the settings by taking indicator cards.

EXPERIMENT 29. Examination of Service Materials Used in a Power Plant

There are many expendable supplies called for in the operation and servicing of a power plant. These supplies should be ordered with great care, for the various compositions entering their fabrication are for the purpose of rendering a specialized service. Examine the materials displayed in the exhibit and make a report on the special items indicated by the instructor, according to the following list.

Note the materials entering the make-up of the fabricated product. Note also the form, an indicator as to its special use. Determine if possible the type of service to be rendered by the material, such as use in connection with the handling of cold or hot water, oils, air, steam, wet, saturated, or superheated, etc. Read carefully all labels attached to the items examined.

1. Packings, metallic, fibrous.

Uses—piston-rod glands; condenser tubes, valve stems, etc.

2. Gaskets, metallic, fabricated.

For pipe flanges, cylinder heads, manhole and handhole covers, etc.

3. Valve seats, metallic, composition.

As used in pumps, line valves, etc.

4. Pipe coverings.

Type of service to be rendered, appearance of cross section, etc.

5. Belting.

Kinds, when and where used. Flat and V-types.

6. Boiler compounds.

Chemical and loading materials in the compound. Special service to be rendered.

7. Fire clay.

Composition, temperature range, where and how used. Mortar, baffles, etc.

8. Firebrick.

Softening temperatures, plastic deformation, temperature range. Furnace walls, baffles, and general shape.

9. Oils and greases.

Type, general characteristics, where used and how introduced in service.

10. Coal.

Kinds and types, conditions of use, mechanical treatment, proximate and ultimate analyses.

11. Others.

EXPERIMENT 30. Technology Report—Small Steam Power Plant

I. OBJECT

This report is intended to cover the student's first technical inspection visit to a typical steam-power plant of less than 1,000 hp.

II. INSTRUCTIONS

1. Live Steam-piping Layout.—Start with the shutoff valve on each boiler and sketch the high-pressure piping to the header and then to each of the main engines or turbines. Indicate the location of each valve and fitting; give the type and purpose of each and the reason for its particular location.

In this layout make a study of the following, and indicate the same on sketch:

Piping	Extra heavy or standard Size Kind of joints Provision for expansion
Fittings	Screwed or flanged Cast iron, malleable iron, or cast steel Name, and how designated
Valves	Gate, globe, check, etc. Brass, iron, or steel Rising or nonrising stem Renewable or stationary seat Bolted or screwed bonnet
Steam separators	Purpose Principle of operation Horizontal or vertical How drained
Steam traps	Type Inlet size How drained
Gaskets	Copper, asbestos composition, rubber, etc.

2. Exhaust Steam, Condensate, and Feed-water Piping.—On a second sketch, indicate the path of the exhaust steam, of the condensate, and of the feed water, from each of the main engines back to the boilers. Observe the general outline given for the previous sketch, and in addition make a study of the following:

Atmospheric exhaust..........	When used Back pressure Disposal of steam
Oil separator.................	Purpose Principle of operation Horizontal or vertical How drained
Make-up or raw-water supply..	Source of water Where admitted How controlled Approximate quantity

3. Main Units.—A description of each of the main boiler and engine or turbine units should be given.

Boilers.................	Number Type and size of each Operating pressure and load on each Fuel used Method of handling fuel How fired
Engines and turbines.....	Number Size and full classification of each Condensing or noncondensing Kind of load Methods of lubrication Procedure in starting

4. Auxiliaries.—A brief description of the type, size and conditions of operation of each of the major auxiliaries should be included in the report.

Condensers.—Surface or jet. Size of steam, water, and air connections. Pressures and temperatures at time of observation. Source of circulating water.

Spray Ponds, Cooling Towers.

Condenser Pumps.—Wet- or dry-air pumps, condensate pumps, and circulating-water pumps should be described.

Feed-water System.—Heaters: open or closed; method of piping; pressures and temperatures at time of observation. Pumps: type, size, and speed. Control: type and method of operation of feed-water regulators and pump governors, if any.

Other Auxiliaries.—Note the type and location of each of the following, if they are provided: superheaters, economizers, forced-draft fans, induceddraft fans, soot blowers, feed-water meters, steam meters, CO_2 recorders, oil purifiers, air compressor.

5. Miscellaneous.—Show by sketches the construction of a globe valve, a gate valve, and a check valve. Define the following: angle valve, reducing valve, blowdown valve, plug cock, gage cock, street ell, pipe coupling, flange union, nipple, bushing. Consult list of fittings and valves (Appendix, page 470).

EXPERIMENT 31. Technology Report—Central Heating Plant

I. OBJECT

This report should give a technical description of a central or district heating system such as the one serving the campus buildings. A description of the heating and ventilating equipment provided in one or more typical buildings served by the system is included.

II. INSTRUCTIONS

1. Boiler Plant.—In studying the central plant, the outline given in Experiment 30 will be followed, with the omission of those parts applying to the prime movers and their immediate auxiliaries. Additional attention should be paid to the return-line vacuum pumps, to the fuel- and ash-handling systems, and to any automatic devices for meeting load fluctuations or otherwise controlling plant operation. Obtain information as to the total number of square feet of radiation served, the amount of fuel used, and the boiler-operating schedule.

2. Distribution System.—Identify and note the size and covering of each pipe or conduit at the main tunnel entrance. Examine and obtain dimensions of tunnel (or buried conduit) construction. Note methods of supporting pipes, provisions for expansion and contraction, and heat insulation. Determine initial pressure and pressure drop in supply mains, noting distances involved.

3. Equipment in Typical Building.—Note kind and size of steam, water, and gas-supply mains, and other service connections. Show by a sketch the location of the steam pressure-reducing valve, indicating the pressures used and the relative sizes of the high-pressure steam main and the low-pressure header system. Sketch or describe thermostatic radiator-control system. Ascertain types, sizes, duty, and methods of control of ventilating fans and motors, air washers and pumps, and heaters for ventilating air. Inspect other mechanical equipment such as water heaters, refrigerating and water-cooling systems, air compressors and exhausters, and elevators. In all cases where instruments are available, note pressures and temperatures.

EXPERIMENT 32. Technology Report—Steam-electric Central Station

I. OBJECT

The following outline will call to the attention of the student some of the main points to be covered in the report on a technical inspection visit to a central station. It is suggested that the style of the report be such as would characterize a descriptive article written for a technical magazine.

II. INSTRUCTIONS

1. Introduction.—Give historical and other general facts you may be able to learn about the station. Comment on station location, type of load, interconnection, and general appearance of the station and grounds.

2. Coal and Ash Handling.—Trace in detail the travel of the coal from railway car to furnace. Give any facts you are able to learn about coal fire protection, coal inventory, checking coal consumption, mixing coals, amount of coal used per day, kinds of coal used, coal sizing and preparation. Describe briefly the ash-disposal system.

3. Boilers and Auxiliaries.—Give general description including type, number, and sizes of boilers and stokers (or pulverized coal burners), also information on kind and intensity of draft, type and drive of fans, stoker drive, furnace-wall construction, feed-water control, blowdown, soot blowing and cleaning, and maintenance of boiler settings. Give type and location of superheaters, economizers, and air preheaters. Discuss methods and apparatus for boiler control, as regards both load and efficiency. What periodic checks are made on boiler performance?

4. Turbines and Generators.—Types and sizes of units, operating pressures and temperatures, and information on bleeder steam extraction and generator air cooling should be given. Inspect oil distribution and cooling system, and obtain information regarding the collection of data for turbine-room log.

5. Condensers and Auxiliaries. Pumps. Exciters.—Describe units, making comments and comparisons. Note types of drive and methods of caring for load variations, also operating pressures and temperatures. Note the means for ensuring reliability of auxiliaries; such provisions as stand-by units, duplicate piping, and steam drives. How is expansion cared for in the piping system?

6. Station Heat Balance.—Trace feed-water heating system from condensate pump to boiler. Discuss methods of providing for changes in load.

7. Electrical Operation.—Determine how the load on the main generators is controlled and how the boiler operators are notified of load changes. Who determines the load to be carried by the station as a whole?

EXPERIMENT 33. Technology Report—Oil-engine Power Plant

I. OBJECT

This report should give a technical description of a diesel-engine power plant. The substance and style of the report are to be such as would characterize a descriptive article written for a technical magazine.

II. INSTRUCTIONS

The following outline is suggested for the report. Notes should be made during the inspection visit, covering these items.

1. General.—Location and general appearance of plant. Historical data. Type and size of building; provision for additional units and for future building additions. Fuel storage and handling facilities. Crane facilities. Interconnection with other plants.

2. Engines.—Number of engines; make and rating of each. Number of cylinders, cylinder dimensions, revolutions per minute, piston speed, two- or four-cycle, solid or air-injection, single- or double-acting, crosshead or trunk-piston type. Kind and size of foundation. Observed pressures and temperatures of fuel oil, lubricating oil, injection air, scavenging air, cooling water and exhaust gases, as indicated by instruments. Disposal of exhaust, appearance of exhaust, exhaust-water heaters or boilers, silencers. Air cleaners and intake mufflers. Method of starting. Kinds of fuel used. Load at time of inspection. Rated and actual brake mean effective pressure.

3. Generators.—Number, make, and full rating of each. Indicated volts, amperes, kilowatts and power factor of load on each generator. Location and drive of exciters.

4. Auxiliary Equipment.—*Cooling Water.*—Number, type, size, capacity, and arrangement of water pumps. Source of raw water and disposal or recooling of heated water. Water treatment. Spray ponds, cooling towers, or heat exchangers. Water pressures and temperatures.

Lubricating Oil.—Number, type, size, capacity, and arrangement of oil pumps. Size and type of oil coolers. Oil pressures and temperatures. Rate of oil consumption. Oil filtering and purifying equipment.

Miscellaneous.—Fire pumps and fire-protection equipment. Shop and repair facilities. Stand-by and emergency units.

5. Operating Data.—Average and peak loads, daily load curve. Average station efficiency or economy. Source of fuel supply, rate of consumption, and approximate cost per barrel.

EXPERIMENT 34. Technology Report—Refrigerating Plant

I. OBJECT

This report should give a technical description of a compression refrigeration plant. The style of the report should be in keeping with the requirements for a good technical magazine article of descriptive nature.

II. INSTRUCTIONS

The following outline is suggested for the notes to be taken by the student during the inspection visit. It is recognized, however, that many installations will not include all the equipment mentioned in the outline.

1. General.—Location and general appearance of plant. Type of service (*i.e.*, cold storage, ice making, air cooling, water cooling, ice-cream making, etc.). Refrigerant used.

2. Compressors.—Type, size, speed, capacity (in tons per 24 hr), and method of drive for each compressor. (Full data on the driving units should be given, including method of governing.) Suction and discharge pressures and temperatures of refrigerant; inlet and outlet temperatures of jacket water. Location and purpose of oil traps.

3. Condensers.—Type (double-pipe, atmospheric, submerged-coil, shell-and-tube), parallel or counterflow, size. Type of joints on condenser and ammonia piping (screwed and soldered, screwed and packed, flanged, welded). Location and size of liquid-ammonia receiver. Source and temperature of cooling water. Recooling or disposal of cooling water.

4. Evaporating Coils.—Locations, pipe sizes, spacings, and arrangement. Methods for controlling total refrigerating load and for controlling refrigerating temperatures. Composition, purpose, and method of using brine.

5. Cold-storage Rooms.—Type of wall construction. Location of ammonia or brine coils. Temperatures maintained and methods of control.

6. Ice-making Facilities.—Can or plate system. Raw or distilled water. Methods and equipment for water treatment or distillation. Provisions for water agitation. Freezing the "core." Ice handling and storage.

7. Other Refrigeration Services.—Description of apparatus for air cooling, drinking-water cooling, ice-cream making, sharp freezing, and other services.

8. Plant Economy and Heat Balance.—Factors affecting plant economy: (1) suction pressure, how controlled and how it affects economy; (2) compressor-discharge pressure, how determined and effect on plant economy; (3) compressor load, speed, and capacity; how governed and how they affect plant economy; (4) exhaust heat utilization from engine-driven units; (5) methods for driving auxiliaries.

Determine, if possible, the average plant performance in terms of pounds of fuel or kilowatt-hours per ton of refrigeration or per ton of ice.

EXPERIMENT 35. Analysis of Automatic Control Devices

I. PREFACE

An automatic control unit consists of three more or less separate functional parts: a detector, an amplifier, and an actuator or operator.

Automatic control is applied to many types of equipment, but the principles are the same whether the quantity to be controlled is the speed of a turbine, the heat supply to an office, the operation of a refrigeration compressor, or the coal supply to a boiler. Speed, temperature, pressure, flow—these four mechanical properties are responsible for the operation of most automatic controls.

The typical control sequence consists of: (1) the *detection* of a change in some mechanical property, such as the pressure in a pipe line; (2) the *amplification* of the effect of this change by some step-up device such as a lever system; and (3) the *actuating* of a corrective or compensating mechanism, as when the lever is used to close a valve.

Form 35.—Analysis of an Automatic Control Unit

Location of unit.............. . ...

Purpose of control.................................

Unit made by..........................Identification.

1. Detector:

   Impulse provided by................................

   Detecting element.............. ....

   Secondary instrument................................

2. Amplifier:

   Class..............................Description.

3. Actuator:

   Class..............................Description.

NOTE: After classifying parts, make a list showing sequence of operations from the time the impulse is received until the corrective function has been accomplished.

TABLE X.—CLASSIFICATION OF AUTOMATIC CONTROL DEVICES

First Unit: The Detector

Sensitive to	Detecting element	Secondary instrument
Pressure	Bourdon tube Diaphragm Bellows Liquid manometer Piezoelectric crystals	Lever mechanism Lever mechanism Lever mechanism Float chamber Galvanometer
Temperature	Expanding solid Expanding fluid Thermocouple Resistance wire	Lever mechanism Bourdon tube Millivoltmeter or potentiometer Wheatstone bridge
Speed	Centrifugal unit Magneto or generator Pump or fan	Lever mechanism Voltmeter Gage or manometer
Electric potential	Voltmeter Galvanometer	
Radiant energy	Photoelectric cell Thermocouple or thermopile Bolometer	
Humidity	Hair, silk, or other fiber Wet-bulb thermometer	Lever mechanism
Position	Selsyn motor	Selsyn motor

Second Unit: The Amplifier

Class	Devices
Electrical	Switch (contact or mercury) and motor Electromagnetic relay Vacuum tube (electrostatic relay) Transformer
Mechanical	Lever system Pilot valve and cylinder or diaphragm Torque amplifier (snubber principle) Fluid pump and motor (Thrustor)

Third Unit: The Actuator

Class	Actuator	Control operation
Electrical	Electromagnet Electric motor	1. Changing a valve or damper opening. 2. Opening or closing an electric circuit. 3. Operating a rheostat.
Mechanical	Lever Wheel and axle Direct pressure, force, or weight Bellows or diaphragm	

Examining each of the functional parts of the control system separately, we see that the **detectors** are in fact instruments for mechanical (or electrical) measurement and that they may be classified under headings which are familiar to the engineer in connection with his instruments for test work.

The **amplifier** or converter furnishes a method for increasing the comparatively weak signal of the detector into a force or an energy stream of sufficient magnitude so that it may be applied to the useful corrective or controlling operation.

The **operator** or **actuator** utilizes the amplified force or energy in the final corrective or control process.

Each of these units may take any one of a large number of forms, and the individual devices will not be described here, but Table X furnishes a basis of classification. This classification is not rigid, and for the simpler controls one or more of the elements may be omitted.

It is to be noted that detectors sensitive to rate of flow of a fluid are not listed in a separate class in Table X. Most rate-of-flow detector elements depend on a difference in pressure, and a pressure-sensitive element is therefore used. (Exceptions to this are the rate-of-flow control units employing either mechanical anemometers or hot-wire anemometers.)

II. INSTRUCTIONS

Make an inspection and analysis of all of the types of automatic control devices to be found in the laboratory and in the campus power and heating plant (or in other plants as designated by instructor). Classification should be made on the basis indicated in Table X. Form 35 is suggested for reporting on each type of control.

CHAPTER IV

OILS, FRICTION, AND LUBRICATION

PART I. PROPERTIES AND TESTS OF LUBRICATING OILS

Experiments 36 to 45

General.—Oils may be classified as animal, vegetable, mineral, and combinations. Some of the more familiar oils grouped under these classifications are:

1. Animal: neat's foot, sperm, tallow, lard, fish.
2. Vegetable: castor, rape, cottonseed, olive, peanut, linseed.
3. Mineral: asphaltic base, mixed base, paraffin base.
4. Combinations: compounds of the above.

Animal and vegetable oils are termed "fixed oils" because, in contradistinction to mineral oils, they cannot be volatilized or distilled without decomposition. These fixed oils possess properties of great adhesion and oiliness or greasiness. They will emulsify readily with water; hence fabrics will wash clean from these oils but not from mineral oils.

Certain of the vegetable oils, when exposed to the air, rapidly absorb oxygen, forming thin, elastic films. They will cause spontaneous combustion when spread thinly over inflammable material. Such oils are called "drying oils," of which the best known is linseed oil. Other vegetable oils show no tendency to form such films, and these are called "nondrying oils."

The following groups of petroleum products may be extracted from the crudes:

Gases and noncondensable hydrocarbons.
Gasolines and volatile distillates.
Kerosene and illuminating oils.
Heavy nonviscous distillates, the fuel oils, etc.
Heavy viscous distillates, steam cylinder and general lubricating oils, etc.
Liquid residues, steam fuel oils, etc.
Semisolid and solid residues.

Mineral oils are further classified as straight cuts and blended oils. **Straight cuts** are oils distilled from a given crude. **Blended oils** are obtained by a mixture of two mineral oils of different characteristics. When the cracking process is used to obtain the maximum yield of gasoline and kerosene from a given crude, some of the heavier components of the crude are broken down into the lighter fractions. Lubricating oils obtained from the residue are generally of inferior quality. High-quality lubricants are obtained from selected crudes which are fractionally distilled with the express purpose of securing the lubricants.

Lubrication.—The subject of lubrication has seen marked development in recent years. Requirements for oils for specific services have received intensive study by the oil industry. Much work has yet to be done to simulate by laboratory tests the actual operating conditions in the field.

In the selection of a lubricant the bearing pressure, speed of moving surface, temperature of operation, and clearance between surfaces furnish the criteria upon which the selection is based.

The purpose of laboratory tests of a given lubricant is to determine its serviceability in reducing wear of moving parts for a given set of the above operating conditions. The attempt in the laboratory tests is to secure in a short period what is to be expected of the lubricant over a long period of time.

The lubricant itself is also subjected to an examination, after test, to determine whether there has developed within the lubricant any change in its physical and chemical characteristics owing to the service rendered.

The following enumerated tests cover the important properties of lubricating oils and serve also as indicators in timing the continued serviceability of an oil:

1. Specific gravity.
2. Flash and fire points.
3. Viscosity.
4. Cloud and pour tests.
5. Carbon residue.
6. Water and sediment.
7. Emulsification.
8. Evaporation test.
9. Free acid.
10. Color number.
11. Penetration (for greases).
12. Friction-machine test.

A number of these properties or physical characteristics of a lubricant are difficult to interpret from a service standpoint. The definitions of the properties and their characteristic tests are presented in the text in the order above enumerated.

Mineral oil, the universal lubricant, possesses the qualities of low shearing resistance and a high resistance to change in physical and chemical properties and furnishes a stable lubricating film.

EXPERIMENT 36. Test of Gravity

I. PREFACE

Specific gravity is defined as the numerical value of the ratio of the weights of equal volumes of oil and distilled water at the standard temperature of 60°F, both corrected for the buoyancy of the air.

Three methods are used in determining gravity: (*a*) by hydrometer; (*b*) by Westphal balance; (*c*) by picnometer.

Use Form 36 to 45 in making report.

II. INSTRUCTIONS

1. Hydrometer Method.—The absolute specific-gravity scale is not commonly used in the oil industry, since all readings are in decimals. Instead, an entirely arbitrary standard scale, called the Baumé scale, is used. For this scale the gravity of water is taken as 10 deg. Liquids heavier than water are designated in degrees below 10, and liquids lighter than water are designated in degrees above 10. Hydrometers that will give an accurate reading to 0.01°Bé may be purchased. A set of such hydrometers are necessary to cover the entire Baumé scale, the range of each being only 5 to 10°. These hydrometers must conform to the A.S.T.M. specifications as to materials and design dimensions. The proper weighted-bulb hydrometer (Fig. 111) with graduated stem is allowed to sink into the oil sample to a level of two smallest scale divisions below that at which it will float and is then released. The depth of immersion is then read as follows:

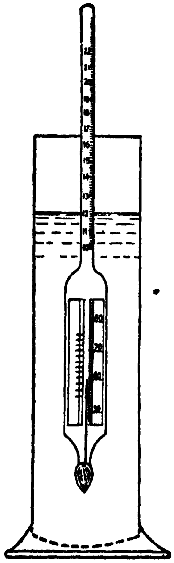


FIG. 111.—Gravity test with hydrometer.

If sample is sufficiently transparent this point shall be determined by placing the eye slightly below the level of the liquid and slowly raising it until the surface of the sample first seen as a distorted ellipse seems to become a straight line cutting the hydrometer scale. For nontransparent oils place the eye slightly above the plane of the surface of the sample and read from the stem the point at which the oil rises above the main surface of the liquid. This reading should be corrected by the amount as determined for a transparent oil having a surface tension similar to that of the sample under test. For routine testing of opaque crude petroleum this correction amounts to a deduction of 0.1°A.P.I. from the observed gravity reading. The standard reading should be made at 60°F. The thermometer, hydrometer, the hydrometer cylinder, and the oil should be near the same temperature. Readings should not be taken until the liquid and hydrometer are free from air bubbles and are at rest. *If the sample tested is not at 60°F a correction must be applied.* For lubricating oils; add or subtract 0.05°Bé (0.10°Bé for kerosene and gasoline) from the observed reading for each degree Fahrenheit according as the observed reading is below or above 60°F.

The oil industry has adopted the Baumé scale specified by the American Petroleum Institute, but some hydrometers are graduated according to the Baumé scale specified by the U. S. Bureau of Standards. Equations for converting readings of either hydrometer to specific gravity are:

American Petroleum Institute (A.P.I.) Scale:

$$\text{Specific gravity at } 60/60°\text{F} = \frac{141.5}{131.5 + \text{degrees A.P.I. at } 60°\text{F}} \tag{17}$$

Bureau of Standards Scale:

$$\text{Specific gravity at } 60/60°\text{F} = \frac{140}{130 + \text{degrees Baumé at } 60°\text{F}} \tag{18}$$

The designation 60/60 means that the specific gravity is obtained by dividing the weight of a certain volume of the oil at 60°F by the weight of an equal volume of water also at 60°F. The A.S.M.E. Code on Density Determinations ("Instruments and Apparatus," Part 16) recommends that "in all cases where practicable" the standard of reference shall be water at 39.2°F.

Duplicate tests should agree to within 0.1°A.P.I. Separate laboratories having different hydrometers, and thermometers should check within 0.5°A.P.I. The precision type of hydrometer will give specific-gravity readings correct to within one point in the third decimal place.

For tables of temperature corrections see *Bulletin American Petroleum Institute;* see also Appendix, Fig. 199, page 488.

When volatile products are cooled or viscous products heated the test cylinder should be placed in a bath maintained at the constant desired temperature during the test. An oil product may lose some of its lighter lower boiling-point constituents, if not handled properly, and hence may introduce an error in the determination of its gravity. A product with initial boiling point below 250°F and a gravity heavier than 70°A.P.I. should be cooled in the original closed container to a temperature of 65°F or lower before pouring into hydrometer cylinder. A product with 70°A.P.I. or lighter should be cooled to 35°F or lower before pouring into hydrometer cylinder.

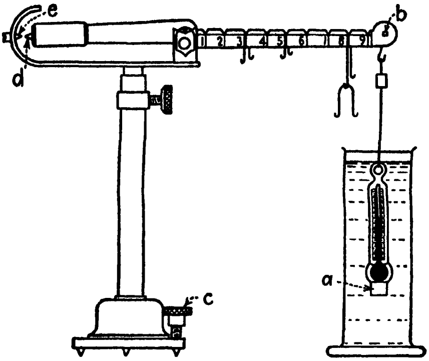


Fig. 112.—Westphal balance.

The thermohydrometer is not used in accurate work on account of inherent defects in design. A separate thermometer is called for: the A.S.T.M. gravity thermometer, range −5 to 215°F, total immersion. There should be a mark on the stem of the hydrometer corresponding to a fixed mark on the scale to provide a check against possible displacement of the scale.

2. Westphal Balance.—This especially constructed instrument (Fig. 112) is much used where a large number of determinations are to be made. Its use makes possible quick and accurate determinations for all nonvolatile liquids.

When the glass plummet *a* is suspended in air from the end of the beam *b* and the instrument is leveled (by means of the base-adjusting screw *c*) the indicating points *d* and *e* should be exactly opposite each other. If the plummet be immersed in distilled water at 60°F (standard temperature), and the largest rider weight be also hung from the plummet hook, it will annul the buoyant effect on the beam and establish equilibrium.

In determining the comparative weight of a given oil sample (having a specific gravity less than that of water or unity), it is obvious that the distilled water-rider weight must be removed from the hook and located at some point on the beam in order to bring the beam to equilibrium. The rider should be moved along the beam toward the knife-edge suspension and located at the first notch reached at which the indicating point *d* falls below that of *e*. To bring the beam into exact balance, weights, which are in decimals ($^1/_{10}$, $^1/_{100}$, and $^1/_{1000}$) of the larger or distilled water-rider weight, are used.

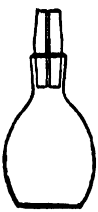


Fig. 113.—Picnometer.

The reading taken from the beam gives the numerical value of the specific gravity direct. Since the beam is graduated in tenths and the rider weights in descending multiples of a tenth of the distilled water-rider weight, the position of each rider read in the order of their decreasing weight gives the numerical value of the specific gravity in the first, second, third, and fourth decimal places. Where two riders fall in the same notch, the smaller may be suspended from the loop of the larger.

A correction of 0.000295 should be made for each variation of 1°F from the standard temperature of 60°F, the correction to be added if the temperature is above 60°F and subtracted if below 60°F (see Fig. 199, page 488). The accuracy of this method is of the same order as that of the precision hydrometer.

3. Picnometer or Specific-gravity Bottle.—The use of the picnometer (Fig. 113) entails much careful and exacting procedure in cleansing and drying the bottle, removing air bubbles, making temperature corrections, etc.

The method of use (A.S.T.M. Standards) is first to weigh the cleaned and dry empty bottle with stopper on a sensitive analytical balance (weight *a*). Then fill bottle with freshly boiled, distilled water at 60°F and weigh (weight b). Take care to expel all air bubbles and fill up to and including capillary tube in stopper. Follow above procedure for oil sample instead of water (weight c).

Substitute in formula to obtain specific gravity of oil sample:

$$\text{Specific gravity} = \frac{c - a}{b - a}. \qquad (19)$$

The bottle contains 25 g when filled with distilled water at 60° temperature.

If accuracy is aimed at in the second and third decimal places, the bottle should be cleansed by use of distilled water and alcohol, or ether. The picnometer furnishes a method of determining the specific gravity of volatile liquids.[1]

If the picnometer has been calibrated, then the calibration curve, plotted between the weight of distilled water and the temperature, will permit the determination of specific gravity with the necessity of but a single weighing.

The gravity of a straight-cut lubricating oil indicates the kind of crude from which it has been distilled. There is, however, no sharp line of separation. The normal division follows:

Kind of oil	°Bé	Specific gravity
Asphaltic base	12 to 29	0.986 to 0.880
Paraffin-asphaltic	29 to 36	0.880 to 0.843
Paraffin	36 to 48	0.843 to 0.786

High-gravity oils (low Baumé), other elements considered, are used for slow-moving and heavy-bearing machinery, low-gravity oils for high-speed and light-bearing pressures. High-gravity oils on burning will deposit a greater amount of residue than light oils. Oils may be compounded to raise the gravity.

EXPERIMENT 37. Flash and Fire Points

I. PREFACE

Specifications designating the flash and fire points of an oil have been handed down through the trade from years ago, and continue to be entered on practically all our varied oil specifications. The flash and fire temperatures were used to check the fire hazard involved in storage and shipping of oils, and this should be their principal use today. The flash point of an oil when considered in connection with a knowledge of the crude from which it was distilled may serve as an indicator of its volatility. The flash point indicates the temperature at which an oil gives off vapors in such proportion that with air they form an inflammable mixture. It should be noted that a lubricating oil will not flash or burn *spontaneously* until it is heated to a temperature considerably above its fire point. Neither the flash nor fire point has any connection whatever with the value of an oil as a lubricant.

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[1] For descriptions of various types of picnometers see *U. S. Bur. Standards, Tech. Paper* 77 and *Sci. Paper* 197.

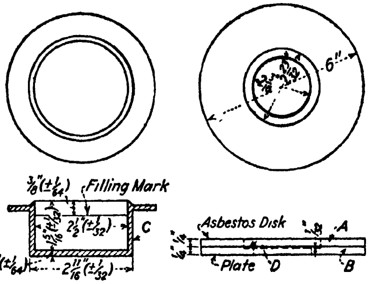

FIG. 114.—Parts of Cleveland open-cup flash tester.

II. APPARATUS AND INSTRUCTIONS

Instruments used to obtain the flash and fire points are of both the open and closed types. Among the closed types the Pensky-Martens (Fig. 114) and the Tag have been approved as standard.

In all open-cup testers the most volatile and inflammable of the vapors escape first, but not in sufficient quantities to be readily ignited.

The rate of heating the sample, size of test flame, distance of test flame from oil surface, kind of test flame (gas, electric), air currents, type of thermometer bulb, depth of immersion, etc., all have their bearing and effect on the temperature of the flash point. Corrections for normal barometric pressure are also to be made where great accuracy is required.

1. Flash Test with Cleveland Open Tester.—This test method is used for all petroleum products except fuel oils and those having an open-cup flash below 175°F. Figure 114 shows the disassembled parts and design dimensions for the complete apparatus. The asbestos plate *A* is placed above the metal plate *B* and the cup *C* placed in the depressed inner ring *D*. The assembly is then placed on a tripod. Oil at room temperature is poured in the cup until the top of the meniscus reaches exactly the depth mark scribed on the inside of the cup.

Heat from any convenient source is applied beneath and centralized relative to the metal plate. The flame is to be so adjusted as to heat the oil at the uniform rate of not more than 30°F per minute rise until a point is reached approximately 100°F below the probable flash point of the oil. Thereafter, a decreased rate of between 9 and 11°F per minute should be used for at least the last 50°F preceding the flash point. The cup should be protected from drafts or excessive radiation by a suitable shield.

The thermometer should be suspended above the cup in such a manner that the bulb will be located ¼ in. from the bottom of the cup and above a point halfway between the center and back of cup.

If the special flash-test thermometer (+20 to 760°F) is not used, correction for emergence of stem should be made. Consult Fig. 17, page 33. The temperature should not be allowed to recede during test but should be kept rising steadily at the rate specified.

The test flame is to be applied at 5-deg intervals. A gas flame $\frac{5}{32}$ in. in diameter is specified. The flame should be drawn across the cup in the plane of its upper edge and in a straight line through the center of the cup and at right angles to the diameter passing through the thermometer. The traverse of the flame should require about 1 sec. The temperature at which the first gas flash takes place at any point on the surface of the oil should be noted and recorded as the temperature of the *flash point.* It is the point at which the oil begins to liberate gas at a sufficient rate to ignite. The true flash must not be confused with the bluish halo that sometimes surrounds the test flame. The room or compartment should be free from air drafts and partly darkened, if possible, so that the flash may be readily discernible.

Fire Point.—If the application of heat is continued (after the flash point has been determined) at the reduced rate of between 9 and 11°F per minute, a temperature will be reached at which gas will be distilled off in sufficient quantity to maintain combustion for a period of 5 sec or more. This temperature is recorded as the fire point.

When the fire point has been reached, remove thermometer at once and smother flame by covering with a metal or asbestos plate. Do not attempt to extinguish flame by blowing.

2. Flash Test with Pensky-Martens Closed Tester.—This test method is intended primarily for determining the flash point of fuel oils. Figure 115 shows a cross section in elevation of the Pensky-Martens tester. *E* is the oil container, which is placed in a metal heating vessel *H*, provided with a mantle *L* in order to protect the heating vessel from loss of heat by radiation. The oil cup *E* is closed by a tightly fitting lid. Through the center of the lid passes a shaft carrying the stirring arrangement, which is worked by means of the handle *J*. In another opening in the cover, a thermometer is fixed. The lid is perforated with four orifices which are left open or covered, as the case may be, by a sliding cover. This cover can be rotated by turning the vertical spindle by means of the milled head *G*. By turning *G*, an opening of the slide can be made to coincide with an orifice in the cover, and simultaneously a very small flame $\frac{5}{32}$ in. in diameter burning at the movable jet *S* is tilted on to the surface of the oil.

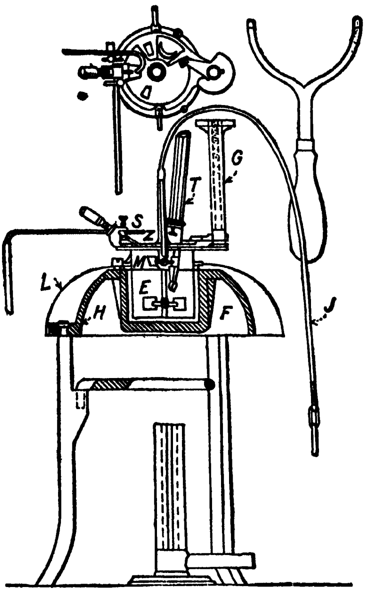

Fig. 115.—Pensky-Martens closed flash tester.

The test is performed by filling the oil cup to a definite mark, fixing the cover, and heating the oil at a rate between 9 and 11°F per minute. The stirrer should make 1 to 2 revolutions per second.

The test flame is applied for each 2°F rise up to 220°F and then for 5°F increments. Record as the flash point the temperature read on the thermometer when a distinct flash in the interior of the cup is noted. The true flash must not be confused with the bluish halo that sometimes surrounds the test flame.

The first application of the test flame shall be made at a temperature at least 30°F below the actual flash point. Application of the test flame shall be made by operating the device controlling the shutter and test-flame burner so that the flame is lowered in ½ sec, left in its lowered position for 1 sec, and quickly raised to its high position. Stirring shall be discontinued during the application of the test flame.

Record the barometric pressure. Duplicate tests should not differ more than 5°F.

III. NOTES

For open cup it is desirable to have a bead of proper dimensions made of some light-colored material to act as a gage in establishing the proper size of the actual bead of flame used.

For open cup, when asphalt or other solid bituminous materials are to be tested, they are heated to 300 to 350°F to make fluid. The procedure to be followed is then the same as above.

For design dimensions of apparatus and special thermometers consult Standards, A.S.T.M.

Corrections: For each inch below 29.92 in. barometric reading add 1.6°F to the flash-point temperature, and for each inch above subtract 1.6°F. Corrections shall be made only in case of dispute. The presence of any gasoline or other fluid used in cleaning should be guarded against.

The flash tests of lubricating oils vary from 300 to 500°F. Ordinarily, the fire point will be 50 to 80°F above the flash point. Since bearings are operated at temperatures below the flash point of even the lightest fractions, the flash point is not a property of practical service value, for it is always below the vaporization point of the oil. Where oil is evaporated or burned as in crankcase lubrication or cylinders of internal-combustion engines, the flash and fire temperatures enter as very important elements in the selection of a proper oil. The use of oils with too low flash points results in the evaporation of the oil from walls and bearings when their normal running heat develops, with consequent rubbing of nonlubricated surfaces. For cylinder lubrication of internal-combustion engines, however, it is preferable not to use oils of too high a flash point, because the oil should evaporate quickly without breaking down into carbon deposits. Consult Table L, page 490, for specifications of various kinds of oils.

EXPERIMENT 38. Determination of Viscosity

I. PREFACE

In general, viscosity is the property of a homogeneous fluid which causes it to offer frictional resistance to motion. This property may be measured in several ways: (1) By the torque required to rotate a cylinder or cup in the liquid; (2) by the time required for a sphere (usually a steel ball) to fall through the liquid; (3) by the time required for the liquid to flow through a capillary or a short tube. The dimensions of the instrument and the quantity of liquid must of course be specified in each case, and hence we have the various standardized viscosimeters[1] such as the Saybolt and Saybolt-Furol oil viscosimeters used in America, the Redwood used in England, and the Engler used in Germany.

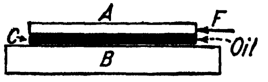

FIG. 116.

To arrive at a proper conception of viscosity, consider plate *A*, Fig. 116, as sliding over plate *B* wth an oil film *C* separating the plates. The force *F* required to move *A* with a velocity *v* is expressed by the formula

$$F = \frac{\mu a v}{d}, \tag{20}$$

from which

$$\mu = \frac{F/a}{v/d} = \frac{\text{shearing stress}}{\text{rate of shear}}, \tag{21}$$

where μ = absolute viscosity, resistance offered by the oil to shearing.
a = area of surface of plate *A*.
v = velocity of plate *A*.
d = thickness of oil film.

The **numerical value of μ**, the absolute viscosity, depends on the units used in Eq. (20). In the metric system the unit is the poise (F = 1 dyne, a = 1 sq cm, d = 1 cm, v = 1 cm per sec). A smaller metric unit, the centipoise, is more often used. Mechanical engineers prefer to use English units, but unfortunately there is no accepted name for the English unit of absolute viscosity. Hence the tables usually give viscosities in centipoises (see Table XIX, page 233, also Fig. 138 and Table XXXIX).

In the calculation of Reynolds number and in many flow equations (see Chap. VII), consistent English units must be used. The A.S.M.E. Fluid Meters Report gives the dimensions of μ in English units as "poundal-seconds per square foot" or the identical "pounds per foot-second." When this unit is used, ρ is the weight-density in pounds per cubic foot, as given in most engineering tables (see Experiments 74 and 75).

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[1] In the American oil industry these instruments have usually been called "viscosimeters," but the A.S.M.E. Code has adopted the form "viscometer."

Kinematic viscosity ν is the absolute viscosity divided by the density. The metric units are the stoke and the centistoke. When μ is in pounds per foot-second it is divided by the ordinary weight-density (specific weight) to obtain ν.

Table XI gives the conversion factors for changing from centipoises and centistokes to the English units.

TABLE XI.—CONVERSION FACTORS FOR VISCOSITY

Multiply the absolute viscosity μ in centipoises by	0.000672	to obtain poundal-sec per sq ft
	0.000672	to obtain lb per ft-sec
	2.420	to obtain lb per ft-hr
	0.0000209	to obtain slugs per ft-sec
Multiply the kinematic viscosity ν in centistokes by	0.001078	to obtain sq ft per sec

As a check on the use of this table, the absolute viscosity of air at room temperature and atmospheric pressure (0.075 lb per cu ft) is given in Table XIX as 0.018 centipoise. Its absolute viscosity is then $0.018 \times 0.000672 = 0.0000121$ lb per ft-sec, or

$$0.018 \times 2.42 = 0.0435 \text{ lb per ft-hr},$$

or $0.018 \times 0.0000209 = 0.000000376$ slugs per ft-sec (check by Fig. 138). The kinematic viscosity of air is

$$0.0000121/0.075 = 0.0001613 \text{ sq ft per sec},$$

or $0.0001613/0.001078 = 0.15$ centistoke. (See also calculation of Reynolds number, Chap. VII, page 259.)

For converting from time of flow in seconds t, as determined experimentally by the Saybolt viscosimeter, to absolute or kinematic viscosity, the following equations may be used:

$$\text{Absolute viscosity, lb per ft-sec} = \mu = At\rho - \frac{B\rho}{t}. \tag{22}$$

$$\text{Kinematic viscosity, sq ft per sec} = \nu = \frac{\mu}{\rho} = At - \frac{B}{t} \tag{23}$$

For the Saybolt standard universal viscosimeter, $A = 0.00000237$ and $B = 0.00194$.

Viscosities of various oils and other liquids are given in Table XL, page 469, and in the Appendix Tables L, LI, and

LII. Viscosities of gases and vapors are given in Table XIX, page 233 and in Fig. 138.

Occasionally the term "specific viscosity" is used. Since the specific viscosity is the ratio of the absolute viscosity to the absolute viscosity of water, the specific viscosity is practically equal to the viscosity in centipoises.

A great many viscosimeters have been devised and used rather extensively. The Saybolt instruments have been standardized by the A.S.T.M. and the U. S. Bureau of Standards, and these instruments are recognized by the Codes of the various American engineering societies. A condensed description of some of the other viscosimeters is given in Table XII.

TABLE XII.—OTHER TYPES OF VISCOSIMETERS

Name	Principle of operation	Used for	Limitations
Gardner-Holt	Falling sphere	Paints	Limited to liquids more viscous than castor oil
MacMichael...	Rotating cup, torsional pendulum	Oils and viscous liquids	Basic calibration is difficult
Stormer......	Rotating cylinder	Clean liquids	Basic calibration is difficult
Ubbelohde. ..	Flow from short tube	Gasoline and low-viscosity liquids	Possible damage to tube; variations in temperature
Redwood.. ..	Flow from short tube	Oils (English standard)	Possible damage to tube; variations in temperature
Engler........	Flow from short tube	Oils (German standard)	Possible damage to tube; variations in temperature
Barbey........	Flow from annular tube	Oils (French standard)	Range from 7 to 500 poises; accuracy questionable

The A.S.M.E. Test Code, "Instruments and Apparatus," Part 17, "Determination of the Viscosity of Liquids," describes and gives instructions for using several types of viscosimeters, but most engineering tests are made with the Saybolt instruments.

A plot on the A.S.T.M. viscosity-temperature chart (Appendix, Fig. 200) will approximate a straight line, excepting for very thin oils having a Saybolt standard viscosity of less than 150 sec or a

Saybolt-Furol of less than 50 sec. By plotting such a graph it is possible to determine the viscosity of an oil at any temperature if the viscosities at two other temperatures are accurately known.

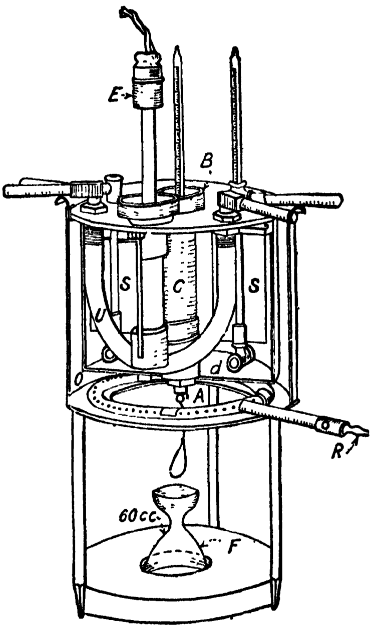

Fig. 117.—Saybolt universal viscosimeter.

The term "viscosity index" recently introduced in the oil trade refers to the slope of the temperature-viscosity curve.

If, therefore, one point and the **viscosity index** are known, the viscosity of the oil at any other temperature may be found.

II. APPARATUS

The Saybolt Universal Viscosimeter.—The instrument sanctioned by the A.S.T.M. as standard, and the one now accepted and most universally employed by the oil trade, is known as the Saybolt standard universal viscosimeter.

Since the rate at which a fluid will flow through an aperture increases as the internal friction of the fluid decreases, we can compare oils or determine the variation in the same oil for different temperatures by means of the orifice principle.

The Saybolt instrument embodies the orifice principle, and the viscosity as determined by its use is defined as the numerical value in seconds of the time for the flow of 60 cu cm of oil through the instrument and is recorded as Saybolt standard viscosity. It is to be understood that there are viscosimeters other than the Saybolt which give very satisfactory results; to reduce their readings in terms of the accepted Saybolt standard, certain factors must be employed (see Fig. 201, page 489). Viscosity scales of all industrial viscosimeters using the efflux principle are purely arbitrary scales.

The instrument (Fig. 117), as built at present, permits heating of bath by steam, hot water, gas flame, or electric-immersion heater. The steam or hot water *U* may also be used as a conveyor of cold water to reduce bath temperature. The A.S.T.M. description of the instrument follows:

The Saybolt universal viscosimeter is made entirely of metal. The standard oil tube *C* (Fig. 117) is fitted at the top with an overflow cup *B*, and the tube is surrounded by a bath. At the bottom of the standard oil tube is a small outlet tube through which the oil to be tested flows into a receiving flask *F*, the capacity of which, filled to the mark on its neck, is 60 cu cm. The lower end of the outlet tube is enclosed by a larger tube, which when stopped by a cork *A* acts as a closed air chamber and prevents the flow of oil through the outlet tube until the cork is removed and the test started (see Fig. 118).

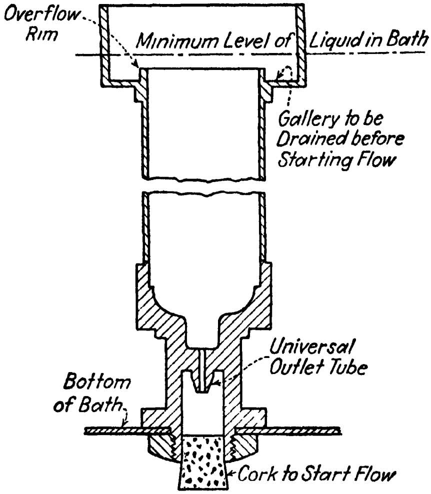
Overflow Rim
Minimum Level of Liquid in Bath
Gallery to be Drained before Starting Flow
Universal Outlet Tube
Bottom of Bath
Cork to Start Flow

FIG. 118.—Saybolt standard oil tube.

Directions for use are as follows: Level up instrument, fill the bath vessel (allow for expansion) with pale engine oil of about 350 to 400°F flash point. Place turntable in position and insert bath thermometer. Start heating and periodically give turntable several quarter-turns by means of the handles. Insert the cork stopper into the lower end of the air chamber at the bottom of the standard oil tube sufficiently far to prevent the escape of air but not to touch the small outlet tube.

Heat the oil to be tested, outside the viscosimeter, to not more than 3° above the temperature at which the viscosity is to be determined, and pour through a 60-mesh strainer into charging cylinder until oil ceases to overflow into the gallery or cup surrounding the top of the cylinder. Use the oil-tube thermometer as a stirrer and bring oil in tube to uniform and test temperature desired; then remove thermometer and by means of a pipette draw off surplus oil in the overflow cup until oil ceases to flow into it over the upper edge of the tube. This ensures a positive volume and starting head. The time of passage of the bottom of the oil meniscus across the 60-cu-cm mark on graduate is taken as the upper time limit.

When oil and bath have been held at their prescribed temperatures for 1 min, snap out the cork plug at base of tube and by means of a stop watch record time in seconds for flow of 60 cu cm of oil into the narrow-necked graduate. This reading in seconds is reported as the viscosity in seconds, Saybolt standard.

Results should be reported to the nearest 0.1 sec for viscosity values below 200 sec and the nearest whole second for values 200 sec or above.

In tests at 100 and 130°F, the bath temperature shall not vary more than 0.1° during test. In tests at 210°F a variation of 0.2° is permitted. Duplicate results should not differ by more than 1 or 2 per cent.

Use Form 36 to 45 in reporting the test. Express results also as absolute viscosity in centipoises.

III. NOTES

For design dimensions of apparatus and special thermometers consult A.S.T.M. Standards.

Since the difference of 0.2°F may make a difference of as much as 1 per cent in Saybolt viscosity, viscosimeters fitted with reliable thermostats are to be preferred. These thermostats may be adjusted to hold the temperature to within a change of 1° or less.

Instrument makers can supply the trade with double-tube viscosimeters with motor stirring devices and accurate thermostats. Duplicate samples may be run in parallel or two different oils checked under identical conditions.

Since unavoidable differences in dimensions between two instruments of the same type will cause greater variations in the results of different laboratories than is allowable, viscosimeters require corrections. The difference between laboratories should not exceed 0.5 per cent.

The U. S. Bureau of Standards is prepared to furnish liquids of known absolute viscosity for the calibration of viscosimeters.

The viscosities of fuel oils, road oils, and other highly viscous materials are determined by the Saybolt-Furol viscosimeter. The test is performed precisely as above outlined for the standard Saybolt instrument, the only difference between the two instruments being that the Furol type is fitted with an orifice that delivers an oil volume at a rate ten times that of the standard orifice. When using the Furol instrument, viscosities are usually obtained at temperatures of 77, 122, and 210°F and are expressed as Saybolt-Furol. In tests on road oils and paving materials additional determinations of Furol viscosity are made at 140 and 180°F. The Saybolt universal viscosimeter is generally used for lubricating oils (and times of flow not less than 32 sec), and the tests are made at the standard temperatures of 100, 130, and 210°F. Room temperatures should be between 68 and 86°F. Determinations should not be made at temperatures below the dew point of the atmosphere surrounding the instrument.

The cork stopper should be inserted not less than ¼ in. or more than ⅜ in. into the end of the air chamber.

While viscosity of oils is usually obtained at the above standard temperatures they may be obtained at any temperature within the range of the instrument, especially at the temperature at which the oil is to be used in service. Light oils, such as straw, dynamo, engine, auto, and cottonseed, may be tested at 70°F. Medium oils, distilled oils such as engine oils, machine oils, and certain cylinder oils, may be tested at 100°F. Heavy gear oils, steam-cylinder or valve oils, airplane-engine oils, road oils, and asphaltic fluxes may be tested at 210°F. Waxes, asphalt, fluxes, paraffin wax, and residues may be tested at 338°F.

For temperatures below 150°F, water may be used as the bath.

The tube and orifice should be cleaned occasionally with a good solvent such as carbon bisulphide to remove any surface film that may adhere to the walls of the tube after continuous use.

The viscosity characteristic of an oil determined by the Saybolt standard viscometer serves not only as an aid to gaging an oil as to its value and place in the field of lubrication, but it also serves in determining the factor of absolute viscosity, a factor that enters into the design of bearings, the flow formula of oils in pipe lines, etc. (see page 258).

The viscosity test does not necessarily prove anything concerning the lubricating value of an oil, as rosin and other loading compounds having no lubricating qualities may constitute a considerable proportion of the whole.

The cause for the drooping viscosity-temperature curve beyond the 150-sec point as stated above may be explained as follows: Gravity pull alone is the actuating force that pushes the oil through the orifice. The resisting force is that offered by the internal resistance of the oil or its viscosity. The time of flow is therefore dependent upon both the density and viscosity. Since the diameter of the efflux tube is large compared with its length, a larger proportion of the energy of flow (particularly for thin oils) is carried away in the issuing stream instead of being used in overcoming the fluid friction inside the tube. The value $B\rho/t$ in the absolute-viscosity formula is the correction factor that must be applied to correct for the descending head on the orifice during flow.

Typical Saybolt viscosities for lubricating oils may be illustrated by the S.A.E. viscosity numbers for motor oils, as given in Table XIII.

Table XIII.—S.A.E. Viscosity Numbers for Crankcase Oils

S.A.E. No.	Viscosity range, Saybolt seconds universal	
	At 130 deg. fahr.	At 210 deg. fahr.
10	90 to 120	....
20	120 to 185	....
30	185 to 255	....
40	255 to	to 75
50	....	75 to 105
60	....	105 to 125
70	....	125 to 150

See also Table L, page 490, for viscosities of oils for various uses.

Oils having too high viscosity will introduce excessive fluid friction at the temperatures and bearing pressures of service. Minimum fluid friction is obtained when the viscosity is such as to just maintain an oil film. It is obvious that for heavy, slow-motion bearings and machinery, an oil of high viscosity is required, and for high-speed and medium to low bearing pressures a low-viscosity oil is required.

Absolute viscosity should be specified in all cases where exactness is important, since of two oils having the same viscosity Saybolt the heavier oil of the two is the more viscous because, being heavier, it will flow more rapidly.

EXPERIMENT 39. Cloud and Pour Tests

I. INSTRUCTIONS

The oil, whether for cloud or pour test, shall first be heated to 150°F and cooled down in air to about 70°F.

Cloud Test.—The cloud test indicates the point at which paraffin wax or other solid substances begin to crystallize out or separate from solution in the oil, when the oil is chilled under definite specified conditions.

Bring the oil sample to at least 25°F above the approximate cloud point, remove any moisture from the sample by some suitable method as by filtration, and then pour the oil to be tested into a round glass jar or bottle of approximately 1¼ in. inside diameter and 4½ to 5 in. high (Fig. 119), to a depth of 2 to 2¼ in.

Insert the special type of thermometer through a tight-fitting cork so that it is held centrally in the jar, with the lower end of the bulb resting on the bottom of the jar. Then place the cold test jar in a metal or glass jacket about 4¼ in. high, having inside diameter ⅜ to ½ in. larger than the outside diameter of the test jar. A disk of felt, cork, or wax, ¼ in. in thickness, is fitted in the bottom of the jacket. A ring *f* is inserted about 1 in. above disk *e* and fitted snugly around test jar and loosely fitted inside of jacket. The test jar is placed in the center of the jacket so that it does not touch the sides at any point. Maintain temperature of cooling bath between 30 and 35°F. Insert this whole apparatus in the refrigerating mixture so that not more than 1 in. of the jacket projects out of the cooling medium. At every drop in temperature of 2°F when near the expected cloud-test point remove the jar from the jacket and inspect, being careful not to disturb the oil by moving the thermometer or otherwise, and return to jacket. This complete operation should require not more than 3 sec.

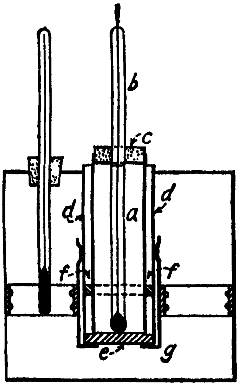

Fig. 119.—Cloud- and pour-test apparatus.

If the cloud point has not been reached at 50°F, the test jar is placed in the second bath which is maintained at a temperature of 0 to +5°F. If the oil does not show a cloud when 20°F is reached, the test jar shall be placed in the third bath maintained at a temperature of −30 to −25°F.

When the lower part of the sample shows a distinct cloudiness or *haze through chilling at the bottom of the test jar, read the* thermometer and record as the cloud-test temperature of the oil.

If oil does not show a cloud when it has been cooled to 15°F, the test jar shall be placed directly in a cooling bath of suitable temperature.

Pour Test.—The pour test indicates the temperature at which a sample of petroleum oil in cylindrical form of specified diameter and length will pour or flow when it is chilled without disturbance under definite specified conditions.

In making this test, the same bottle, quantity of oil, and cooling device are used as for the cloud test, except that the thermometer bulb shall be placed so that the beginning of the capillary shall be ⅛ in. below the surface of the oil.

Heat (without stirring) the oil in the test jar to a temperature of 115°F in a bath maintained at not higher than 118°F. Then cool to 90°F in air or in a water bath at approximately 77°F. Oils on which a pour point below −30°F is expected shall be heated as above with the high cloud- and-pour test thermometer in position, cooled to 60°F and the low cloud- and-pour test thermometer placed in position.

Starting at 20°F above the expected pour point and at each drop in temperature of 5°F, remove bottle from the jacket; incline from vertical just sufficient to ascertain whether there is a movement of the oil in the test jar. The complete operation of removal, testing, and replacement shall require not more than 3 sec.

If oil has not ceased to flow when its temperature has reached 50°F (as in the cloud test), the test jar shall be transferred to the second bath. If then the oil has not ceased to flow when its temperature has reached 20°F, then the test jar shall be placed in the third bath, maintained as in the cloud test. If oils of still lower pour points are tested, baths will be required with successively lower temperature differentials of about 30°. In each case the test jar shall be transferred when the temperature of the oil reaches a point 50°F above the temperature of the new bath. When the oil in the test jar is chilled to the point where it ceases to flow when the jar is inclined to the horizontal and remains there for exactly 5 sec, then the pour point shall be taken as the temperature 5°F above this temperature reading of the solid point.

Care must be taken in handling the test jar, as any disturbance of the spongy network of wax crystals will lead to low and fictitious results.

II. NOTES

A.S.T.M. cloud- and pour-test specifications for lubricating oils require at least three and preferably four cold test baths which must be maintained at the following temperatures:

1. Temperature +30 to +35°F.
2. Temperature + 0 to + 5°F.
3. Temperature −30 to −25°F.
4. Temperature −60 to −55°F.

It is well also to make a test of a sample of the oil without previously heating it. Heating previous to chilling increases crystallizing action and causes oil to congeal at a higher temperature. Agitation during cooling will delay congealing 15 to 25°F.

The pour test should be given consideration when selecting oils for refrigeration equipment or for exposed machinery in cold climates.

Paraffin oils have a pour temperature normally between 25 and 35°F. This may be lowered by continued refining, removing the paraffin. Asphaltic-base oils have a pour temperature of 0°F.

A special procedure for black oils and cylinder stocks follows: If a submitted sample of oil has been previously subjected to a temperature of 115°F or above within a 24-hr period and the history of the sample is not known, then, if possible, the sample should be held for 24 hr before testing. Otherwise, three consecutive readings of the same sample shall check each other. This cold point is recorded as the upper (maximum) pour point.

The low (minimum) pour point shall be determined after heating the sample, while stirring, to 220°F, pouring in the test jar, and cooling to 90°F. The upper and lower values of the pour points shall be reported separately.

The materials used in the freezing mixture may be ice and calcium chloride, or sodium chloride, depending upon the temperature desired in making the tests. For oils congealing or solidifying above 35 to 15°F, a mixture of cracked ice and a small addition of salt, 1 to 20 by volume, may be taken. For temperatures from 15 to −5°F, also use an ice-and-salt mixture, adding about one-third salt. From 0 to −25°F, a mixture of ice and calcium chloride is used. Dry ice is now procurable on the open market and makes an excellent refrigerant for temperatures normally between −25 and −70°F. Dry ice sublimes slowly. In proper containers it loses in weight only from 2 to 8 per cent per 24 hr. Dry ice may be used in connection with acetone or gasoline.

Dry ice may be made from liquid CO_2 by drawing off the liquid from the cylinder through a chamois skin bag. Rapid evaporation will cause solid CO_2 or dry ice to form.

EXPERIMENT 40. Carbon Residue

I. APPARATUS AND INSTRUCTIONS

If oils are heated to a high temperature in the absence of a free supply of air, the greater part of the oil will distill, leaving a carbon residue. The amount deposited is an indicator of the extent to which decomposition will take place in service, especially in the case of oils for internal-combustion engines, for domestic oil fuel, or oils used in the manufacturing of gas.

Conradson's method of determination is the one in standard use. The apparatus (Fig. 120) consists of:

1. **Porcelain crucible, *a*, glazed throughout, or silica crucible; 29 to 31 ml capacity; 46 to 49 mm in rim diameter.**

2. Skidmore iron crucible, *c*, 65 to 82 ml capacity, 60 to 67 mm outside diameter at top and 30 to 32 mm outside diameter of the flat bottom. Crucible to be 37 to 39 mm high with cover without delivery tubes and having vertical opening closed. The horizontal opening of about 6.5 mm to be kept clean.

3. Iron crucible, *b*, formed from spun sheet iron of approximately 0.8 mm in thickness, with cover; 78 to 82 mm in outside diameter at top and 58 to 60 mm in height. At the bottom of this crucible a leveled layer of sand is placed about 25 ml of dry sand, or enough to bring the Skidmore crucible with cover on nearly to the top of the wrought-iron crucible.

4. Triangle, of bare nichrome wire, approximately 13 B & S gage, having an opening small enough to support the bottom of the sheet-iron crucible at the same level as the bottom of the asbestos block or hollow sheet-metal box.

5. Hood of circular sheet-iron, *e*, from 120 to 130 mm in diameter, the height of the lower perpendicular side to be from 50 to 53 mm. The chimney to be 50 to 60 mm in height and from 50 to 56 mm in inside diameter. The cone-shaped part shall be constructed so as to bring the total height of the completed hood to 125 to 130 mm. As a guide for flame height above chimney, a bridge of ⅛ in. iron or nichrome wire is attached to chimney and having a height of 50 mm above top of chimney.

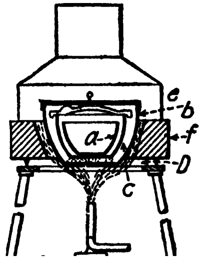

FIG. 120.—Conradson carbon-residue apparatus.

6. Insulator. Asbestos or hollow sheet-iron block or refractory ring 150 to 175 mm in diameter provided with a metal-lined inverted cone-shaped opening through the center 83 mm in diameter at bottom and 89 mm diameter at top.

Procedure.—Place the two glass beads provided, which are about 0.1 in. in diameter in the porcelain crucible, and include their weight in the net weight of the crucible. Then add 10 g of the oil to be tested, accurately weighed and free from moisture and suspended matter.

Place the crucible in the center of the Skidmore crucible. Level the sand in the sheet-iron crucible, and set the Skidmore crucible on it in the exact center of the iron crucible.

Apply covers to both the Skidmore and the iron crucible, the one to the latter fitting loosely to allow free exit to the vapors as formed.

Place the bare nichrome wire triangle on the metal tripod and on it place the asbestos block or hollow sheet-metal box.

Center the sheet-iron crucible in the block with its bottom resting on top of the triangle, and cover the whole with the sheet-iron hood in order to distribute the heat uniformly during the process.

Apply heat with a high strong flame from a Meker-type gas burner so that the preignition period will be 10 ± 1½ min (a shorter time may start the distillation so rapidly as to cause foaming or too high a flame). When smoke appears above the chimney, immediately move or tilt the burner so that the gas flame plays on the sides of the crucible for the purpose of igniting the vapors. Then remove the heat temporarily, and before replacing adjust gas flame so that the ignited vapors burn uniformly with flame above the chimney, but not above the wire bridge. Heat may be increased, if necessary, when the flame does not show above the chimney. The period of burning the vapors shall be 13 ± 1 min.

When the vapors cease to burn and no further blue smoke can be observed, readjust the burner and hold the heat as at the beginning so as to make the bottom and lower part of the sheet-iron crucible a cherry red and maintain for exactly 7 min. The total period of heating should be 30 ± 2 min, which constitutes an additional limitation on the tolerances for the preignition and burning periods.

There should be no difficulty in carrying out the test exactly as directed with the gas burner provided, using city gas (about 550 Btu) with the top of the burner about 2 in. below the bottom of the crucible. The time periods should be observed with whatever burner or gas used.

Remove the burner and allow the apparatus to cool until no smoke appears (about 15 min); then remove the cover of the Skidmore crucible. Remove the porcelain crucible with heated tongs, place in the desiccator, cool, and weigh.

Then {total weight crucible plus beads and 10 g oil at start} minus {weight of crucible plus beads and carbon residue at finish} equals loss of oil in grams. Then 10 g oil minus loss in weight of oil equals carbon residue in grams. Therefore, carbon residue in grams divided by 10 g equals percentage of carbon residue in sample.

Tolerances: Weights of oil sample should be accurate to within 5 mg. Tests should be run in duplicate and repeated if necessary until the percentages of carbon residue differ by not more than 10 per cent from an average.

The entire process should require about ½ hr to complete when heat is properly regulated. The time will depend somewhat upon the kind of oil tested, as a very thin, rather low-flash-point oil will not take so long as a heavy thick, high-flash-point oil.

The carbon residue increases nearly in proportion to the increase in viscosity.

For oils having carbon residues in excess of 5 per cent, such as heavy crudes, residua, heavy fuel oils, and heavy gas-enrichment oils, the procedure as above may encounter difficulties. The sample may boil over, or difficulty may arise in dehydrating. When the sample shows by the usual method of test a carbon residue in excess of 5 per cent, the test should be repeated in duplicate using a 5-g sample.

EXPERIMENT 41. Water and Sediment

I. APPARATUS AND INSTRUCTIONS

Crude oils as mined contain dirt and earth substances, water, salt, and emulsions. All crudes must undergo a settling process to eliminate these contaminants. Lubricating oils when new are presumably free of water and sediment, and these tests in the case of lubricating oils are designed for used oils. These tests are of particular interest in the case of used oils from circulating systems of turbines and engines, including internal-combustion engines, and various types of fuel oils.

Centrifuge Method of Testing.—This method is not entirely satisfactory because the amount of water obtained is nearly always lower than the actual water content. This method is in such prevalent use, however, as to receive recognition for commercial approximations.

The difficulties of obtaining representative samples for this test are usually great. Hence, the method of sampling cannot receive too much attention. The portion tested must be thoroughly representative of the original oil.

The preferred form of centrifuge shall have a diameter of 15 to 17 in. (tip to tip of centrifuge tubes) and be revolved at a speed of about 1,500 rpm. Any other diameter centrifuge shall be run at an equivalent tip speed. The slope of the tubes shall be optional (see Fig. 121).

Procedure.—Fifty cubic centimeters of 90 per cent benzol shall be measured into each of two centrifuge tubes, and 50 cu cm of the oil to be tested shall then be added to each. The centrifuge tubes shall be tightly stoppered and shall be shaken vigorously until the contents are thoroughly mixed. The temperature of the bath shall be maintained at 120°F and the centrifuge tubes shall be immersed therein to the 100-cu-cm mark for 10 min. The two centrifuge tubes shall then be placed in the centrifuge on opposite sides and shall be whirled at the rated rpm, or the equivalent, for 10 min. The combined volume of water and sediment at the bottom of each tube shall be read and recorded as such, estimating to 0.1 cu cm if necessary. The centrifuge tubes shall then be replaced in the centrifuge, again whirled for 10 min as before and removed for reading the volume of water and sediment as before. This operation shall be repeated until the combined volume of water and sediment in each tube remains constant for three consecutive readings. In general, not more than four whirlings shall be required. Duplicate readings should not differ by more than 0.2 cu cm.

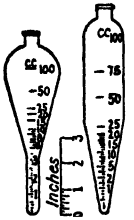

Fig. 121.—Centrifuge tubes.

The combined total volume of water and sediment should be read on each tube. The sum of the two readings shall be recorded as percentage of water and sediment—centrifuge method.

II. NOTES

The automotive type of engine is the worst offender in regard to contamination of lubricant. In the crankcases of these engines the oil becomes diluted with fuel. Carbon from the cylinder walls and underside of the piston flakes off and falls into the crankcase. Sand and dirt get into the engine through the carburetor and breather. Metal from abrasion and corrosion from water are always present. Sludge, gums, and deposits form from the above contaminations until the oil has to be removed. The amount of sediment found on testing used motor oils or the heavier oils from the circulating system of diesel engines is in a measure an indication of the carbon-residue condition of the oil.

MISCELLANEOUS OIL TESTS

EXPERIMENT 42. Emulsification

I. INSTRUCTIONS

Equal parts of distilled water and oil are churned together in a shaking machine for 1 hr at the rate of about 350 rpm. Upon completion of test, the oil and water should separate quickly. At the end of 1 hr there should be no film of emulsion between the oil and the water. Any tendency of oil to hang in water indicates lack of proper refining. The emulsification test is an indicator of purity and degree of refining. Commercial refining involves treatment of oil with acid for the purpose of removing undesirable hydrocarbons, the acid being washed out afterward by water. Free acid in oils increases emulsibility. Alkaline compounds will often cause emulsification. In fact, any impurity or adulterant is conducive to the formation of emulsions. Oils after being in use may be checked for further usefulness, as above.

To determine percentage of emulsification, the following method is suggested.[1] To 40 cu cm of distilled water, contained in a clean 100-cu-cm graduate, add 40 cu cm of the oil sample to be tested and raise in a water bath to a standard temperature of 130°F. The content of the graduate is then poured into a 250-cu-cm beaker which is immersed in a water bath of 130°F and stirred by an electric stirrer 100 rpm for 5 min (hexagonal stirrer disk on motor-driven stem, $\frac{3}{32}$ in. thick and $\frac{7}{8}$ in. across the flats). Emulsion is then returned to graduate and allowed to stand for 30 min in water bath 130°F, after which readings for emulsifying properties are taken. Also readings are to be taken after standing for 24 hr at room temperature of 70°F. Percentage is recorded as permanent emulsion. The "emulsion value" is calculated from the percentage of oil separated as compared with the amount of oil taken, after 24 hr standing.

II. NOTES

Heating emulsions in many cases serves to break them down, since change in the ratio of density of water to oil increases as the temperature rises and causes quicker and more thorough separation. The factor having the greatest influence on separation due to heating is the viscosity. The decrease in viscosity of oil upon rise in temperature is much greater than that for water for the same range of temperature, and the water in seeking its lower level has not the resistance to meet in pushing aside the globules of oil that would be present if the mixture were at a lower temperature. The lower the viscosity, the more rapid the separation. From the above facts, the oil having the steepest viscosity-temperature curve, other conditions remaining constant, has the greatest ability to separate from a temporary emulsion.

In some cases, as in a Westinghouse steam-engine crankcase, experience has shown that a mixture of oil and water is the most satisfactory lubricant. This mixture is more easily distributed to the working parts than is oil alone, and this furnishes better lubrication An oil too heavy will not properly mix with the water, and an oil too light in body will not film properly over the wearing parts. An oil free from any emulsifying action must be used.

Low-grade mineral and compounded steam-cylinder oils are difficult to separate from exhaust steam.

For certain classes of lubrication, oils are compounded in order to make them emulsify. This emulsifying property of compounding oils is utilized where the application of water is used to prevent overheating of bearing. The water that reaches the bearing, instead of washing away the oil film, produces a rich oily emulsion or lather of high lubricating value.

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[1] Consult *J. Soc. Chem. Ind.*, 1915.

Steam-cylinder oils are as a rule compounded, although a straight mineral oil may be used where service time is not a controlling factor. Compounded steam-cylinder oils are loaded with wool grease, tallow, or lard oils, or a combination of these highly viscous materials. In the presence of high temperature and moisture, a loaded steam-cylinder oil will readily emulsify, the emulsified part adhering to the wet cylinder walls and thereby furnishing lubrication, whereas a straight mineral oil will be washed off the walls by the entrained moisture. Commercial loaded steam-cylinder oils contain from 5 to 10 per cent of compounding.

For certain uses, mineral oils are compounded with vegetable oils, such as compounding with castor oil.

EXPERIMENT 43. Evaporation Test

This is a test as applied especially to lightweight oils such as high-speed spindle oils. More than 4 per cent of volatile is not permissible.[1]

EXPERIMENT 44. Free Acid

Since most oils are refined by use of acids, traces of the acid may be found at times in commercial oils where the utmost care has not been used in the refining. Again, fatty oils used in compounding introduce free fatty acids.

To detect presence of acid shake vigorously together equal volumes of oil and warm distilled water. Permit oil to separate from water. Pour oil off and test the water with litmus paper for acid reaction. The litmus paper may be made extremely sensitive if it is previously prepared, by holding in nitric acid fumes until turned partly red, and then dried. A very small percentage of acid present may be detected by its turning the litmus a darker red. This is not a true test, as some acids are not soluble in water.[2]

Animal and vegetable oils break up readily under heat with the formation of acids. These acids are inactive with metals but cause emulsions and soaps which dissolve in the oil and cause it to become thick. Mineral acids are harmful to bearing surfaces and, in addition, aid emulsification where water is present. Straight mineral oils will generally develop acidity when subjected to high temperatures in the presence of air and moisture. The harm is, however, in the readiness to emulsify.

EXPERIMENT 45. Color Number

One of the characteristics that refiners strive for in finished lubricants is good color. Color is not in itself of great value except as a means of identification. Color has nothing to do with lubricating value. Color is merely an identification mark. Oils may be colored to suit the likes or dislikes of the oil trade. Either light or dark oils may be distilled from the same crude. In finishing, it is possible to give a normally light oil a dark color or vice versa.

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[1] For details of test, see *Proc.* A.S.T.M.

[2] For quantitative analysis of oils for free acid, see *Proc.*, A.S.T.M., or *Bur. Mines*, *Tech. Paper* 323*a*, p. 80.

The oil trade uses the Union colorimeter in checking color value. Light is passed through a standard bottle filled with the oil to be tested and the color of this light compared with colored glass slides corresponding to shades of standardized chromate solutions. When the Saybolt chromometer is used, the light transmitted through the oil contained in a glass tube is compared with that transmitted through a color standard glass and the color number taken from a standard chart.

Good oils may discolor and become darker but remain clear after cooling, if raised in temperature too near the flash point. Poor oils will get black showing decomposition and likely the formation of acids.

Form 36 to 45.—Results of Lubricating-oil Tests

General data	
Kind of oil........................	Made by........................
Trade name and grade........................	Color No............
Date of test........................	Laboratory No..............
Observers..	
Test results	
1. Baumé gravity at...deg. fahr...	4. Cloud point, deg. fahr..........
Baumé gravity at 60 deg. fahr...	Pour point, deg. fahr...........
Instrument used...............	5. Carbon residue, per cent........
Specific gravity at ... deg. fahr.	6. Emulsification.................
Specific gravity at 60 deg. fahr...	....
Instrument used...............	7. Water and sediment by centrifuge, per cent...............
2. Flash point, deg. fahr...........	
Fire point, deg. fahr............	8. Free acid.....................
Instrument used...............	....
3. Viscosity at...deg. fahr.........	9. Color number.................
Viscosity at...deg. fahr.........	....
Viscosity at...deg. fahr.........	
Viscosity at...deg. fahr.........	
Above viscosities are expressed in seconds, Saybolt.............	

Date................ Observers:.

Oils that contain more or less air, if held at a temperature above 140°F, have a tendency to oxidize, owing to the intimate contact of air and oil when subjected to bearing pressures. Heavy oils have the deeper colors. Oils should not be darker than their viscosity warrants, as such a condition is evidence of incomplete or improper refining. Highly filtered oils are paler than other oils of corresponding viscosity, but the color of oils can be lightened by acid treatment.

Oils should be free from turbidity which might indicate the presence of water, paraffin, glue, etc. Cylinder oils, if free from tar, are green rather than black in color.

PART II. PROPERTIES AND TESTS OF GREASES

(Semifluid Oils)

EXPERIMENT 46. Hardness and Melting Point

I. PREFACE

In all types of greases from fluid to solid, the lubricating properties of the grease are entirely determined by the lubricating properties of the unsaponified oils and hydrocarbon oils contained in them. The question then is, why manufacture any greases at all? The reason is that there are no satisfactory commercial lubricants of hydrocarbon- or fatty-oil origin which are sufficiently thick or have the characteristics required for the lubrication of all types of transmissions; hence it has been necessary to find a means of artificially thickening available oils to the desired degree. Various classes of grease (thickened oils) are manufactured to meet this need.

Greases are usually made from the combination of a mineral oil with a soap formed by saponifying an animal or a vegetable oil with an alkali. The soaps are formed from any combination of horse fat, tallow, lard, stearic acid, oleic acid, lard oil, cottonseed oil, etc. These soaps possess the remarkable property of acting as a sponge for the absorption of hydrocarbon oils to such an extent as to hide completely the presence of the oils.

Petroleum oil products are practically insoluble in water. They have a characteristic greasy touch and low surface tension. The low surface tension permits the oil to spread readily and penetrate pores of dry substances. When certain oils are boiled in solution of caustic alkalies one class remains unaffected, the other breaks down into fatty acid and glycerin and is then said to be saponified. Petroleum oils are unaffected by boiling caustic. Vegetable and animal oils are readily saponified.

The quality and the characteristics of a grease depend upon the characteristics of the oil used, the alkali used for saponification, the amount of water incorporated, and the method of manufacture. It is claimed that the manner of manufacture of any grease has a greater influence upon its quality and characteristics than does its chemical composition. Unless grease is skillfully and scientifically mixed and the proper heats used in the mixing process, separation of oil and soap constituents will result. The thickness of the grease is determined by the quantity and viscosity of the hydrocarbon oil added to the soap.

Greases are classified according to the kind of soap used as the base, *viz.*, lime-base greases, soda-base greases, and special greases.

The lime-soap-base greases (most extensively used as cup greases) are emulsions of mineral oil and water, stabilized by the soap base. The presence of an excess amount of water in a lime-base grease will cause a dull or cloudy appearance. If the water content is not sufficient, the grease will show a granular composition. A lime-base grease should be bright and have a buttery consistency.

The stability of the emulsion should be such as to permit of no oil separation from the grease while in storage. If the water of the emulsion evaporates, owing say to high operating temperature, the mixture, on cooling, will not possess its original characteristics, owing to separation of constituents.

These greases, which have a comparatively fluid consistency, will meet average operating conditions under normal temperatures and pressure. If high temperatures are present, a soda-base grease, which possesses a heavier body, should be used.

Soda-base greases are as a rule formed from an intimate mixture of soap and oil, free from any water content. They are not therefore emulsions. Since caustic-soda soap greases are partially soluble in water, these greases should not be used where water contact is present. These greases will in most cases show an acid reaction, but, if moisture is absent, no corrosion will result, as the acid is quite inactive.

Aside from the standard line of straight-run greases with the lime- and soda-soap bases, special greases for abnormal service conditions are manufactured. These special greases are the lime- or soda-soap greases loaded with such materials as talc, soapstone, mica, graphite, etc. These loading materials are soft and nonabrasive in character. They function but to a limited degree as a lubricant themselves. They do assist lubrication, however, by filling in the uneven or porous structure of the bearing surface, making the same smooth and thus aiding in the formation and maintenance of the necessary lubricating film. These greases serve effectively where extremely high-pressure and high-temperature conditions prevail.

The adaptability of a grease as a lubricant for a given service is based upon the governing characteristics of hardness and melting point.

The quality of a grease is determined from the properties of: (1) consistency or hardness; (2) melting point; (3) solubility in water; (4) texture; (5) acidity; (6) quality of mineral oil.

No satisfactory standard has been devised for a number of these tests.

II. APPARATUS AND INSTRUCTIONS

The A.S.T.M. has designed a special cone-shaped plunger apparatus (Fig. 122) as a means for measuring the hardness of a grease. The depth of penetration of the cone, as indicated by a pointer read in millimeters, gives the hardness factor.

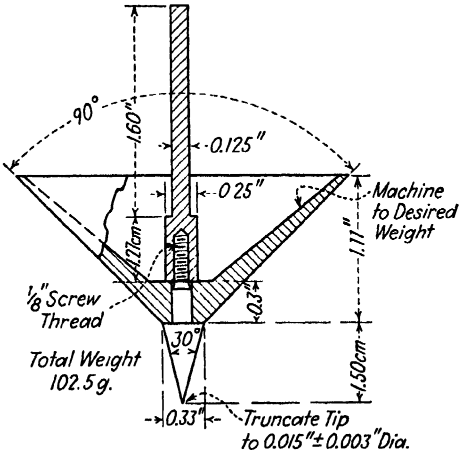

FIG. 122.—Cone for grease test.

The cone is built to the dimensions indicated, with a hardened stainless-steel tip and a brass body.

Total moving weight in the test shall be 150 g. If the plunger weighs 47.5 g, the cone or the cone plus an attached weight must weigh 102.5 g. It is in general desirable to make the cone as light as possible so that it will have a wide range of usefulness, and to add a weight to give the desired total for the grease tested. If the conditions of the tests are not specified, the penetration of a grease shall be understood to refer to the results of this test made at 77°F.

The cone is inserted in a standard penetrometer (A.S.T.M.). The cone is lowered until the tip just touches the grease surface. Watching the shadow of the tip aids in accurately setting the cone. The scale is then set to zero and the plunger released suddenly and kept released for 5 sec. The penetration is read from the scale in tenths of a millimeter.

The average of five tests on each sample shall be reported as the consistency if the mean deviation does not exceed 3 per cent; otherwise, the average of 10 determinations shall constitute the consistency.

To determine the melting point by "drop test," a small amount of grease is smeared on the bulb of a thermometer which is then suspended in an empty test tube. The tube is slowly heated, and the temperature reading corresponding to the point at which the grease melts and drops off the bulb is recorded as the melting point of the grease.

Texture is determined by observation whether the grease is smooth, butterlike, stringy, fibrous, or crystalline.

Acidity is that of the mineral oil and is very undesirable.

III. NOTES

The effective lubrication of a machine depends upon the adaptability of the lubricant to the actual operating conditions. Greases are used instead of oil under conditions of service where the high pressure, high temperature, continuous shock, intermittent operation, or slow-speed operation furnishes conditions where a normal oil film would be destroyed. Greases furnish seals against entry of dust and grit into the polished surface of balls, rolls, or shaft. They are free from splash, drip, and leakage and therefore prevent waste.

Most trade-marked greases are characteristically scented to hide the odor of fatty oils or pitches with which they are made, and also to furnish the trade an easy means of identifying brands of grease by the sense of smell.

As to color, greases may be dyed to give the desired tint, or the color may be determined by that of the hydrocarbon oil used. A new consistometer instrument has been devised by the U. S. Bureau of Standards[1] for application to greases and oils at low temperatures.

[1] *Research Paper* 188.

PART III. FRICTION AND LUBRICATION

General.—The word "lubrication" is derived from the Latin *lubricus* which means slippery. The idea is to introduce something between sliding surfaces which is slippery.

The frictional resistance offered by the contact surfaces of two bodies sliding or rolling on each other, and dependent upon the area of contact, the velocity and character of surfaces, and the normal force, often presents a serious problem to the engineer. In many cases it is desirable to have this resistance of contact a maximum, as in the case of the tractive effort between the driver of a locomotive and the steel rail (tracks are often sanded) or between a belt and a pulley (belts are treated with resinous dressing). In the case of shafts or pivots turning in journal bearings or cups, however, the reverse, or minimum, resistance is demanded in order to keep the loss of useful work at a minimum. The latter condition is secured by introducing a lubricant between the surfaces, separating the metal-to-metal contact. Thus, there is developed only the resistance due to shear in the film of the lubricant. The work done in overcoming this resistance appears in the form of heat and is dissipated to the air; or to water for a water-cooled bearing.

EXPERIMENT 47. Friction Test of Lubricating Oil

I. PREFACE

Many variable elements enter in to affect the type of oil to be used for a given service, such as pressure of rubbing surfaces, provisions for heat radiation, clearance between surfaces, kind of bearing material, as well as character of bearing surfaces, variation of speed or load, quality of oil, and frequency and type of oil feeding. The proper lubricant is the one giving least energy losses under service conditions, provided it does not develop undesirable characteristics such as acidity and gumming.

The need of proper lubrication is evident when it is known that our power plants show friction losses of from 5 to 40 per cent, and further that good over poor material and methods of lubrication may result in a saving of 10 or more per cent in power by the reduction of these friction losses. Friction, however, means not only loss of power but rapid depreciation, continuous repairs, and consequent shutdown of machinery.

The function of a lubricating oil is to supply a film to prevent sliding or rolling surfaces from coming into contact with each other, thus reducing wear, keeping down temperature rise of surfaces, and reducing energy losses. Experimental results show that, at the running temperature, an oil of minimum viscosity that will maintain a sufficiently thick film to prevent contact of rubbing surfaces will show the minimum coefficient of friction.

The fluid friction present in an oil causes resistance to motion, for, when a bearing floats on an oil film, the principal friction is due to the internal friction of the fluid itself. If the fluid friction of an oil is so little that the movement of the oil is faster than that of the moving surface, the oil will flow out and leave the surface exposed. If the opposite condition prevails, however, the oil being so heavy as to lag behind the moving surface, then the oil will offer resistance and more power will be required to overcome the total resistance to motion. Since heat thins oils, it is obvious that the oil that will maintain the mean between the above mentioned two conditions for the running temperature of the surface will be the one having the proper body.

To determine the relative efficiencies of oils as to their actual lubrication value, a friction test is usually made.

II. APPARATUS AND INSTRUCTIONS

A number of machines that employ various methods of measuring the friction work have been designed to determine the coefficient of friction.

1. The Thurston Pendulum Machine.—This is perhaps the most universally employed. It is direct reading at any load and speed. Referring to Fig. 123 the machine consists of a weighted pendulum *A* which is free to turn on the mounted overhung shaft *B*. Variation of speed is in part secured through the step pulleys *C* and in addition any variation through the driving machine itself. The bearing pressure is obtained by means of the coiled compression spring *D*, confined in the cylinder *E* of the pendulum, which puts equal loads on both halves of the pendulum bearing *F*. This pressure may be varied by means of a base-adjusting nut *G* and is read directly from the pendulum scale.

To obtain the pressure in pounds per square inch, divide this pressure by the projected area of the bearing. The area of both upper and lower bearings should be considered. Usually this value may be read off the pendulum scale located directly opposite the total spring-pressure reading. The resistance to motion (total friction) turns the weighed pendulum through an angle which is read from the graduated arc.

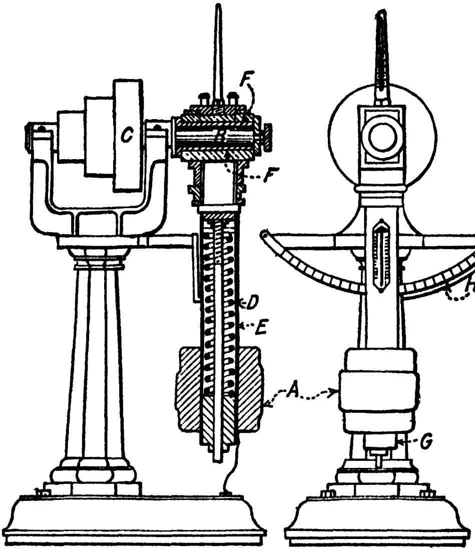

Fig. 123.—Thurston oil-friction tester.

Coefficient of friction

$$= \frac{\text{total friction}}{\text{total spring pressure} + \text{weight of pendulum}}$$

These machines are made in several different sizes for light-pressure, high-speed, and for heavy-pressure, low-speed determinations.

Let r = radius of shaft, inches,
l = length of bearing, inches.

Then,

$2(2r)l$ = projected area, square inches.

If F represents the total friction,
P represents the total pressure,
f represents the coefficient of friction,
R represents effective radius of pendulum,
W represents weight of pendulum,

T represents compression of spring, pounds,
θ represents pendulum arc,
we may write

$$P = 2T + W.$$

Since $f = F/P$ by definition, we have

$$Fr = Pfr = f(2T + W)r = WR \sin \theta,$$

from which

$$f = \frac{WR \sin \theta}{rP} \qquad (24)$$

$WR \sin \theta/r$ is read on the fixed arc, which varies in value as θ varies, since W, R, and r remain constant. Check for accuracy. To determine WR adjust the pendulum to an exact horizontal position and weigh the downward thrust on a platform scale. Determine W by weighing the complete pendulum. Secure also measurement of l and r.

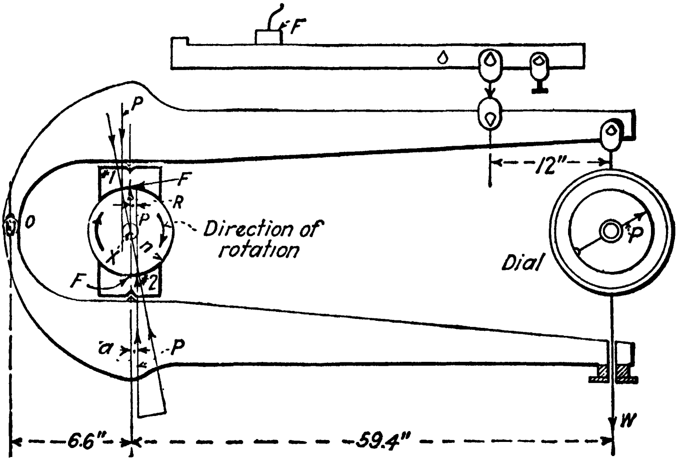

FIG. 124.—Oil-friction test machine employing "nutcracker" leverage system.

2. The Olsen and Riehle Machines.—These machines (Fig. 124) consist of a rotating shaft revolving between two brass boxes 1 and 2 which may be clamped together under pressure, which pressure is transmitted through a set of levers and read upon a graduated scale beam. The tendency to rotate is also measured by means of a lever system and read either from a scale beam or from a dynamometer spring scale.

In the use of the above types of machines, the conditions of bearing pressures and speeds at which the oil is being tested should be as near as obtainable to the same values as the oil is to be subjected to when in use. The machines permit of bearing pressures as high as 750 lb per sq in. projected area, and speeds of 500 ft per min, which are as high as are ordinarily reached. Where the motion is exceedingly slow, pressures ten times the above have been reached, in practice.

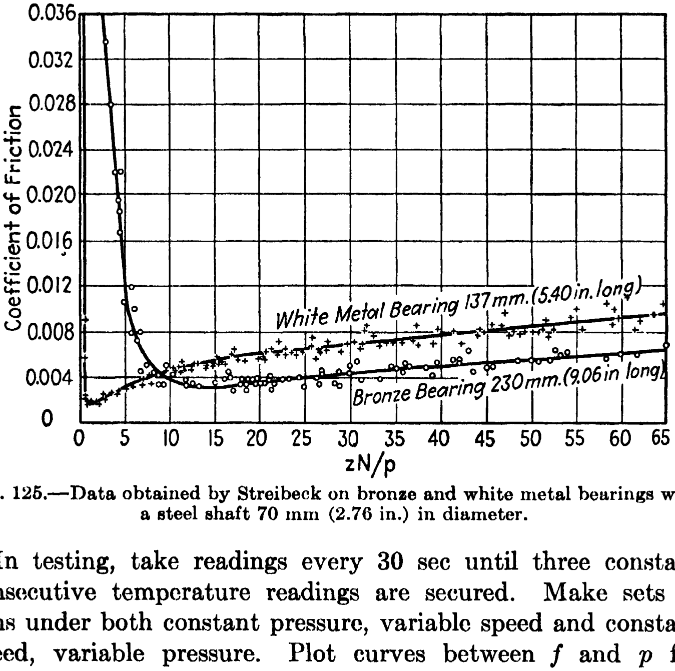

FIG. 125.—Data obtained by Streibeck on bronze and white metal bearings with a steel shaft 70 mm (2.76 in.) in diameter.

In testing, take readings every 30 sec until three constant consecutive temperature readings are secured. Make sets of runs under both constant pressure, variable speed and constant speed, variable pressure. Plot curves between f and p for constant speed and f and revolutions per minute for constant pressure.

Plot typical performance curves as shown in Fig. 125. Determine z from the viscosity-temperature curve and plot also curve between f as ordinate and **friction factor** $\frac{zN}{p}$ as abscissas for both constant-load and constant-speed runs.

Let z = absolute viscosity at the given temperature.
N = number of rpm.
p = load, pounds per square inch.

The lower the value of f the less the power lost in shearing the oil film. This point (the lowest on the above curves) may be near the rupture point of the oil film and a dangerous operating point. Between runs, cool off bearings and cleanse thoroughly. Some machines have hollow bearings through which cold water may be made to pass to expedite cooling.

III. NOTES

Were it not for the destructive effect that heat has upon lubricating oils, they could be used almost indefinitely if kept clean. But heat is the greatest enemy of hydrocarbon lubricating oils, causing the destruction of their requisite properties. The degree of heat decomposition that occurs in service during a given period is determined by the relative heat-resisting properties of oil, *i.e.*, upon its resistance to chemical change into nonlubricating products.

Friction of lubricated surfaces is determined by the nature of the lubricant rather than by that of the solids or bearing metals themselves.

Synthetic lignum vitæ bearings are now being manufactured by the General Electric Company, using water as the lubricant. Rubber bearings, water lubricated, are a special product of the Goodyear Rubber Company.

Speed of minimum friction is the rate of motion at which, if varied from either faster or slower, the friction of a bearing increases. With perfect lubrication, it is believed to be between 100 and 150 ft per min.

Heavy (viscous) oil may fail to distribute evenly in close-fitting bearings and allow some parts to become injured while other parts are flooded.

The total friction of a given steam engine remains practically constant under all loads and should not exceed 5 to 8 per cent of the full-load capacity of the engine. It increases with steam pressure from 30 to 70 lb after which only slight increase is noted. Increased speed generally but not always increases the total friction.

Rosin oil is cheap and its specific gravity high; therefore, it is sometimes used to adulterate lubricating oil. When it has been considerably refined, it can be added in large quantities without much chance of detection except by a chemist; but, since it has no lubricating qualities, the action of the mixture will be unsatisfactory.

Cottonseed and corn oils are sometimes used in lubricating oils but have merit only as cooling elements for thread cutting and the like. Their high specific gravity (92 to 93 per cent) makes the mineral oil to which they are added deceptive when too much dependence is placed on viscosity.

Allowable pressure per square inch ranges from 10 to 30 lb for high-speed shafts in babbitted bearings to 100 lb for large slow-speed shafts. Engine crankpins often sustain 1,000 to 3,000 lb per sq in.; crossheads, 1,000 to 1,200; main bearings, 800 to 900; guides, etc., 25 to 60; line shaft, 10 to 30 lb per sq in.

From recent investigations fostered by the A.S.M.E. on lubricants under pressure, the following has been reported:

Pressures reaching 57,000 lb per sq in. and temperatures as high as 284°F have been used.

Mineral oils show a greater increase of viscosity than the fixed oils over a moderate range of pressure. Paraffin case oils show a tendency to solidify under a certain temperature-pressure relation. In general, the temperature coefficient of viscosity is considerably increased under moderately high pressures. These findings have a particular significance in the study of oiliness phenomena.

Methods have now been devised to predict very closely bearing performances if the physical characteristics of the oil and the clearance and pressure of the bearing are known.

Form 47.—Friction Test of Lubricant

General description

Kind of oil........................Trade name..................
Made by........................ ....
Laboratory No........................Barometer............in. Hg
Room temperature............deg. Fahr.

Friction-test results

Machine used........................Method of feed................
Bearing, type........................Journal, type..................
Bearing, size........................Journal, size..................

Run number........................	1	2	3	4	5	6
R.p.m........................						
Surface velocity, ft. per min............						
Total pressure........................						
Pressure, lb. per sq. in. projected area.....						
Temperature, deg. Fahr................						
Friction, total........................						
Coefficient of friction..................						

Date................ Observers:.

When a condition of moisture is present, tallow or lard oil is added to break down the surface tension of an oil and increase its adhesive properties.

For an increase in penetration qualities or in bearings with small clearances lard oil is often added to the lubricant.

Our present knowledge confirms the statement that no compounding of a lubricant will reduce the friction unless this addition changes the viscosity, *i.e.*, different oils of the same viscosity at a given temperature will generate the same amount of friction through shearing at identical service conditions where the pressures are below the point of film rupture.

Extreme Pressure Lubricants

Made up as follows from standard lubricants:

1. Sulphur + mineral oil with or without saponifiable oil.

2. Saponifiable oil treated + sulphur chloride + added mineral oil.

3. Mineral oil containing lead soap with or without additions of sulphur.

4. Oil with other compounds of chlorine and of sulphur.

The oil adulterants form a type of lubricating film preventing scoring, the mineral oil acting as the vehicle for the adulterants and as a cooling agent.

The factors that determine the ability of an extreme pressure lubricant to perform its duty satisfactorily are: Its load-carrying capacity, tendency to prevent wear, corrosive property, and its stability.

IV. REFERENCES ON LUBRICANTS AND LUBRICATION

1. General Discussion on Lubrication, A.S.M.E., 1,200 pp.

2. A.S.T.M., "Standard on Petroleum Products and Lubricants," prepared by Committee D-2, 375 pp.

3. U. S. Government Specifications for Lubricants and Liquid Fuels and Methods for Testing, *Bur. Mines, Tech. Paper* 323*a*.

CHAPTER V

THE TRANSFER OF HEAT

I. THE GENERAL PROBLEM

Heat is our primary source of power; it is applied to useful purpose in almost every manufacturing industry, and it must be added to or taken from nature's atmosphere in order to provide for human comfort. The control of heat flow and heat potential (temperature) is therefore one of the major activities of the mechanical engineer. In spite of this fact, the study of heat flow as a specialized subject is comparatively new. Even today we have no convenient and accurate heat-flow meters and no generally accepted unit of heat flow.

As compared with other countries, the low cost of our primary fuels in the United States furnishes one reason why we have not paid more attention to the engineering study of heat flow. Another good reason is found in the difficulties involved in heat control and heat measurement. Our best heat insulators are far from perfect, and heat flows and leaks in every direction in the most complicated paths.

Fortunately most of the conditions of engineering practice may be closely approximated by simple cases, based primarily on the fundamental laws of conduction, convection, and radiation. Most heat-exchange equipment may be studied on the basis of the simple case of the heat flow between two fluids separated by a solid wall. Tests of complete equipment and such aids as dimensional analysis are also valuable.

II. ANALYSIS OF THE HEAT-FLOW PROCESS

Perhaps the most common and easiest method of describing or picturing heat transfer is to consider it as a "flow" process and to compare it with two other examples of flow which occur in engineering work, *viz.*, the flow of water (and other fluids) and the flow of electricity.

Using the first analogy, it is convenient to picture graphically the "**temperature gradient.**" The "hydraulic gradient," with which the student is already familiar, pictures the losses in "pressure head" caused by successive resistances in the path of flow. Since temperature is a measure of "thermal pressure" or "heat head," the temperature gradient pictures the similar losses or drops in pressure along the heat-flow path. Thus, in the simple and frequently occurring case of a wall of metal, brick, or other solid material separating two fluids, the temperature gradient will appear somewhat as shown in Fig. 126. The temperatures T_1 and T_2 are those of the two fluids, and T_a and T_b the two surface temperatures of the wall.

Turning to the electrical analogy, the various temperature drops are heat "potential drops" and correspond to the "*IR* drops" in the electrical circuit. Moreover, if the resistances can be evaluated, the calculations involved in the heat circuit are very similar to those of the electric circuit (see Solution of Engineering Problems, Sec. IV, page 206). The resistances encountered are essentially of two types: (1) surface resistance and (2) conductor resistance (see Fig. 127). In dealing with the surface resistance to the flow of heat between a fluid and a solid, the term **"film effect"** is sometimes used. This term, however, refers essentially to heat transfer by convection. The friction between the fluid and the wall tends to retard the motion of a thin "film" of the fluid and thus to reduce the transmission of heat by convection. In many cases, however, radiation plays an even greater part than convection, and the "emissivity" or "absorbing power" of the materials then plays the major part in determining the surface resistance (see Table XIV page 198, and also pages 203 and 207).

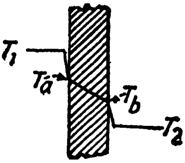

FIG. 126.—Temperature gradient—simple wall.

Temperature Gradient of Common Heat-transfer Processes.—If the wall in Fig. 126 were that of a common **refrigerator,** it would be made of such materials and of such thickness that the temperature on the outside surface T_a would be very little lower than that of the surrounding air T_1. Similarly, the inside wall temperature T_b would be very nearly that of the inside air T_2. The conductivity of the wall material is the all-important thing in this case, and the surface finish of the walls and the movement of the air over them would have little effect on the heat loss. *These conditions obtain in most problems of heat insulation.*

On the other hand, suppose this wall to be that of a steam or hot-water **radiator.** Common experience tells us that the temperature T_b of the outside surface of this wall is almost as high as that of the steam or water T_1. Nearly all of the drop is occasioned by a single surface resistance, from T_b to T_2, and any method by which this resistance can be reduced will increase the heat flow. In other words, those conditions should be created at this surface which will increase the amount of heat transmitted by convection and radiation. It will make little difference whether the wall of the radiator be of cast iron, a rather poor conductor, or of copper, which is a very good conductor.

As a third example, and one that involves certain complications, let us consider the case of a **boiler tube.** In actual service the wall of a boiler tube is not a simple steel wall but rather a steel wall with a coating of soot and ash on one side and a coating of scale on the other (Fig. 127). These materials introduce additional temperature drops. The surface resistances are large, the formation of steam on the inner surface of the tube tending to increase the resistance at that surface. *The means at hand for accelerating the heat flow in the case of a boiler tube are then about as follows:* (*a*) Keep both surfaces clean, thereby reducing the temperature drop across the additional layer of material at either surface of the steel tube. That is, keep the conductivity of the composite solid wall separating the two fluids as high as possible. (*b*) Increase the velocity of each fluid, thus increasing heat transfer by convection. (*c*) Place the tube where it can "see" the fire, thus increasing the amount of heat absorbed by radiation. Increasing the CO_2 content of the flue gases will also increase the radiant heat absorbed, since CO_2 is not "transparent" to heat waves as are the constituents of air. On the contrary, CO_2 (and also water vapor) will absorb and radiate heat in much the same manner as do solids and liquids. Increasing the content of CO_2 is then somewhat similar to subjecting all the heating surface to the radiant effect of a luminous flame.

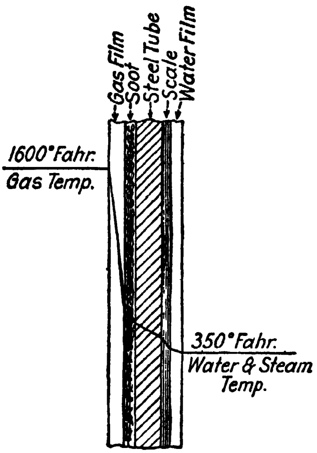

Fig. 127.—Temperature gradient—boiler tube, compound wall.

A similar analysis of other heat-transfer processes will suggest in each case certain methods or means by which the heat transfer may be accelerated or retarded, as the case may require. This information will then furnish a basis for the design of the apparatus or for improvements in existing designs. Operating conditions and maintenance or cleaning schedules will also be dictated, as in the case of the boiler tube discussed above. A thorough knowledge of the known laws governing each type of heat transfer is necessary for such analyses, and the following partial summaries will be found useful.

III. LAWS OF RADIATION, CONDUCTION, AND CONVECTION

1. Radiation. *a. Nature of Radiation.*—In the spectrum of radiant energy, usually called the "ether spectrum," heat radiation occupies a comparatively narrow band. Similar electromagnetic radiation of longer wave length includes the radio broadcast band, whereas the range of shorter wave lengths is from visible light, ultraviolet, and X rays to gamma rays and cosmic rays.

Heat radiation may be absorbed, reflected, or transmitted, but it is converted into ordinary sensible heat only when it is absorbed. Most solids and liquids are opaque to nonluminous heat radiation, and they either absorb or reflect it. Polished metals are the best reflectors. Most gases are transparent to radiant heat, but at furnace temperatures carbon dioxide and water vapor are good absorbers and good radiators. Heat radiation is a surface phenomenon and depends on the final surface condition but is not directly related to color or other optical properties.

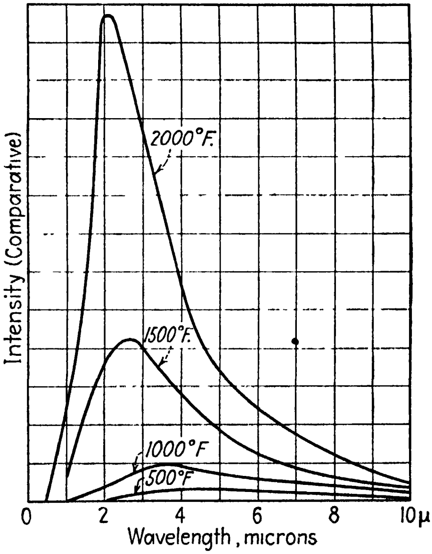

FIG. 128.—Distribution of radiant energy according to wave length, as defined by the Planck law.

b. Radiation Laws.—A body that absorbs all incident radiation is known as a **black body**, and the common laws of heat radiation apply primarily to such a body. The *Stefan-Boltzmann law* states that the total radiant energy emitted or absorbed by a black body is proportional to the fourth power of the absolute temperature. The equation that gives the distribution of the energy intensity according to wave length is known as the *Planck law* (Fig. 128), and the *Wièn displacement law* defines the relation between the temperature of a body and the wave length of its maximum emission. *Kirchhoff's law* states that for the same temperatures the emissive and absorptive powers of a body are equal. The *Lambert cosine law* states that the unit intensity of radiation

TABLE XIV.—RADIATION FACTORS OR EMISSIVITIES
For the determination of factor F_1 in Eq. (25)

Class	Surfaces	Percentage of black-body radiation		
		At 50–100°F	At 1000°F	Solar radiation
1	A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99	0.97 to 0.99	0.97 to 0.99
2	Black nonmetallic surfaces such as asphalt, carbon, slate, paint, paper.........	0.90 to 0.98	0.90 to 0.98	0.85 to 0.98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)........	0.85 to 0.95	0.75 to 0.90	0.65 to 0.80
4	Yellow and buff brick and stone, firebrick, fire clay...	0.85 to 0.95	0.70 to 0.85	0.50 to 0.70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash................	0.85 to 0.95	0.60 to 0.75	0.3 to 0.5
6	Window glass..............	0.90 to 0.95	....	Transparent
7	Bright aluminum paint; gilt or bronze paint...........	0.4 to 0.6	....	0.3 to 0.5
8	Dull brass, copper, or aluminum; galvanized steel; polished iron..............	0.2 to 0.3	0.3 to 0.5	0.4 to 0.65
9	Polished brass, copper, monel metal...................	0.02 to 0.05	0.05 to 0.15	0.3 to 0.5
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04	0.05 to 0.10	0.10 to 0.40

NOTES: *a.* Factors apply to either radiation or absorption.

b. For a small body in a large enclosure, use the emissivity of the small body only: $F_1 = e_1$.

c. For rectangles or disks, either parallel or perpendicular and with a common side, use the product of the emissivities: $F_1 = e_1 \times e_2$.

d. For infinite parallel planes, concentric cylinders or large enclosed bodies, use both emissivities in the equation:

$$F_1 = \frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1}.$$

received upon a surface that is not normal to the ray is proportional to the cosine of the angle of incidence.

For most practical calculations of either radiation or absorption a modified form of the Stefan-Boltzmann law is used:

$$q \doteq 0.173\, F_1 F_2 A \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right], \qquad (25)$$

where T_1 and T_2 are the temperatures of the radiating and receiving surfaces in degrees Fahrenheit absolute, q is the total radiant heat exchange in Btu per hour, and A is the area of the smaller body in square feet. The total spherical radiation from a black body is 0.173 Btu per sq ft per hr. The factor F_1, from Table XIV accounts for departures from complete blackness. The factor F_2 from Table XV accounts for the orientation or configuration of the surfaces, or the extent to which one body "sees" the other.

TABLE XV.—ORIENTATION FACTOR F_2, FOR USE IN EQ. (25)

Case no.	Description	Values of factor F_2
1	Infinite parallel planes, completely enclosed bodies or infinite concentric cylinders	1.0
2	Parallel or equal squares or disks of width D:	
	a. Distance between $= D$	0.2
	b. Distance between $= 0.5D$	0 4
	c. Distance between $= 0.25D$	0.6
	d. Distance between $= 0.12D$	0.8
3	Adjacent squares with common side	0.2

c. Radiation of Solids, Liquids, Gases, and Particles.—Most nonmetallic **solids and liquids** radiate about as *gray* bodies with an emissivity of 90 to 95 per cent of that of a black body, *i.e.*, their energy distribution curve is similar to that defined by the Planck law (Fig. 128) with ordinates 90 to 95 per cent as great. Polished metal surfaces have very low emissivities, and they may not follow exactly the cosine law. For slotted or corrugated surfaces of any materials, the emissivity increases with the relative depth of slot, and the *envelope* rather than the developed area should be used in computing the radiation. This applies to such objects as air-cooled cylinders and multicolumn steam radiators.

Nonluminous radiating gases radiate mainly in certain narrow ranges of wave lengths, and the total radiation depends on the product of the partial pressure of the gas and the thickness of the body of gas, *i.e.*, the diameter of the gas passage. The common radiating gases are water vapor, carbon dioxide, carbon monoxide, and the hydrocarbons. **Luminous flames** contain incandescent particles and have a high emissivity. For accurate calculation of radiant-flame heat transfer, special empirical methods are necessary.

Absorption of *solar radiation* and heat lost by *night radiation* of a body in open space at night are two problems of some engineering importance which do not lend themselves to simple treatment by the radiation laws. The radiant heat absorbed by a body in direct sunlight depends upon the latitude as well as upon the orientation of the surface and its material and color. For U. S. latitudes in midsummer the maximum intensity of **solar heat** reaching a normal surface is about 300 Btu per hr, and from 9 A.M. to 3 P.M. it is above 200 Btu per hr. A black surface may absorb 90 per cent of this radiation, but a very light-colored surface may absorb only about 40 per cent of it. Ordinary glass will transmit over 80 per cent of solar radiation, but it is almost opaque to nonluminous radiation.

Terrestrial radiation, from the warmed earth to space on a clear night, is about 40 Btu per hr.

2. Conduction.—Thinking in terms of our everyday experience in dealing with heat conduction through metal objects or through solid walls of metal or other materials, we readily conclude that the amount of heat transferred varies directly with the temperature difference and inversely with the length of path. We recognize, moreover, that there is a great difference in the conductivity of the common materials and that materials naturally fall into two classes, heat "conductors" and heat "insulators."

Formulating a law from experimental results, these relations may be expressed mathematically as follows:

$$q = \frac{kA(T_1 - T_2)}{L}. \qquad (26)$$

The very close parallel of electrical conduction is apparent from an examination of each factor in this equation. In fact, keeping in mind that conductance is the reciprocal of resistance and that temperature is a measure of heat "potential," this equation may be seen to correspond to Ohm's law:

$$q = \frac{T_1 - T_2}{L/kA} \quad \text{and} \quad I = \frac{E}{R} = \frac{E_1 - E_2}{L/kA}. \qquad (27)$$

In either case the resistance may be expressed as L/kA, or the conductance as kA/L, where k is the "conductivity" or unit conductance and is the reciprocal of the unit resistance. The total resistance of the path, furthermore, is directly proportional to its length and inversely proportional to its area. It is to be noted that q is a unit of heat current or flow rather than of heat quantity and it is expressed in Btu per hour. The conductivity k may be defined as "the heat transmitted by conduction in Btu per square foot per hour per degree Fahrenheit difference in temperature per inch of thickness (or length of path)." This factor is not a constant but varies with the temperature, being in most cases a straight-line function of temperature (compare electrical conductivity). When values of k are selected for substitution in the heat-conduction equation, care must be taken that the **mean conductivity** over the given temperature range is used. In Table XVI are given the mean conductivities of certain common materials.

The conductivity of engineering materials is closely related to their structure and physical properties. The heat conductivities of metals are of the same order as their electrical conductivities. The heat conductivity of a gas is proportional to the product of its viscosity and its specific heat at constant volume. Pipe coverings and insulating wallboards are effective largely because of the presence of air cells in their structure, and their conductivity therefore increases with temperature. Such fibrous insulators as are manufactured from wood pulp, straw, paper, cornstalks, sugar cane, cork, etc., all have a minimum conductivity at room temperature of about 0.27. This is much increased if the material is not dry. Heat insulators for use at temperatures above the atmospheric range (as for instance steam-pipe coverings and insulating bricks) have higher conductivities both because the solid inorganic materials that must be used to resist combustion are better conductors and because the entrained air has a higher conductivity at the higher temperatures (see Table XVI).

The above discussion refers primarily to steady-state conduction through homogeneous solids. For nonhomogeneous materials or for two materials in surface contact, an over-all conductance must be obtained from available experimental data. When heat storage is involved, as in the case of heavy walls of furnaces or buildings, or in the heating and cooling of large objects, the process must be studied in terms of short time intervals, and an accurate solution involves integration of differential equations.

3. Convection.—In a large number of technical heat-transfer processes, convection plays a more important part than either radiation or conduction.

TABLE XVI.—CONDUCTIVITIES, SPECIFIC HEATS, AND DENSITIES

For other properties of solids and liquids see Table XL, page 469
For other properties of gases see Table XIX, page 233
All properties at 70° unless otherwise specified.

Material	Density, ρ = lb per cu ft	Specific heat, c_p = Btu per lb per deg	Conductivity, k = Btu per hr per °F per in. thickness
Metals:			
Aluminum	167	0.22	1,415
Brass	530	0.09	750
Copper	550	0.095	2,690
Iron, gray cast	445	0.13	340
Iron, pure	490	0.107	440
Lead	710	0.030	239
Magnesium	108	0.24	1,100
Nickel	550	0.105	406
Platinum	1,335	0.032	482
Silver	656	0.056	2,950
Steel, structural	485	0.11	400
Tin	455	0.055	430
Zinc	445	0.093	770
Building Materials and Other Solids:			
Brick, common	110	0.22	5.0
Brick, hard (face brick)	130	0.24	9.2
Coal, solid	100	0.3	
Concrete, stone	140	0.16	12
Firebrick	140	0.26	9
Glass, window	160	0.16	6
Ice	57	0.52	
Plaster	95	0.25	3.3
Stone, building	150	0.2	12.5
Wood, oak	45	0.55	1.2
Wood, pine or hemlock	30	0.65	0.8
Insulating Materials:			
Asbestos millboard	60	0.2	1.0
Corkboard	10	0.45	0.30
Fiberboard	15	0.5	0.33
Fiber (or hair) blanket	6	0.5	0.27
Mineral wool	10	0.2	0.27
Insulating furnace brick	28	0.2	0.8
85% magnesia pipe covering	15	0.2	0.5
Liquids:			
Alcohol, ethyl	49	0.56	1.26
Gasoline, motor	46	0.55	1
Glycerin	79	0.58	2.0
Kerosene	50	0.5	1.05
Mercury	845	0.033	50
Oil, lubricating	53	0.5	1.15
Water, 70°	62.2	1.0	4.0
Water, 210°	59.9	1.0	5.0
Gases (at atmospheric pressure):			
Air, 70°	0.075	0.24	0.16
Air, 200°	0.0602	0.242	0.20
Air, 1000°	0.0272	0.25	0.33
Carbon dioxide	0.0724	0.20	0.095
Hydrogen	0.0052	3.4	1.1
Steam 212°	0.037	0.49	0.15

The common engineering definition of convection is that it is "the carrying of heat by a fluid in motion." If this motion is a gravity flow due to the change of density with temperature, as when the warmed air rises along the column of a steam radiator, the process is called "natural" or "free" convection. *Free convection* is usually considered as taking place in a large body of unstirred fluid. If, on the other hand, the main body of fluid is in motion due to outside forces, as in the case of a fluid flowing in a pipe or duct, the heat-transfer process is called *forced convection.* Of course a large number of border-line cases exist in which the velocity of the main stream is small and the gravity circulation plays a more or less important part.

Although these definitions are roughly descriptive of the convection process, they do not describe the exact mechanism nor do they give any clue to the quantitative calculation of the heat transfer. To provide such an explanation and a method of calculation, there has been evolved the **film theory of convection.** We know from common experience that there is a great difference in temperature between the main body of fluid and the confining wall, in the case of a bare steam pipe in air, for instance. This large drop in temperature (potential) implies a high surface resistance and leads to the idea that a thin film of slowly moving fluid adheres to the wall surface and interferes with the free flow of heat. The theory states that the fluid flow within this film is viscous or streamline in character (see page 258) and that the heat transfer across the film then takes place entirely by conduction. Since the fluid (air, for example) has a low conductivity, there results a large drop in temperature across the film. At the outer boundary of the film the flow changes rather suddenly from viscous to turbulent, and the temperature of the main stream is reached.

Setting up an equation for the calculation of this convection heat flow, we resort to the simple form:

$$q = hA(T_1 - T_2), \tag{28}$$

where q = heat flowing, Btu per hour.
h = "surface coefficient" for the conditions of the problem (see Table XVII).
A = mean area of the heat path, square feet.
T_1 and T_2 = temperatures of the surface of the wall and of the main body of the fluid, T_1 being the higher temperature, degrees Fahrenheit.

TABLE XVII.—APPROXIMATE VALUES OF FILM OR SURFACE COEFFICIENTS
(h = Btu per hour per square foot per degree Fahrenheit difference between the mean temperature of the fluid and that of the wall surface)

Part I. Forced Convection (Radiation *not* included)

Fluid	Flow path	Mean fluid velocity through free area			
		2 fps	5 fps	10 fps	50 fps
Air.....	Over flat surfaces	.....	2	3	10
Air.....	Through 1-in. pipe	.....	1.8	3.2	11.5
Air.....	Through 3-in. pipe	.....	1.5	2.6	9.2
Air.....	Through 12-in. duct	.....	1.2	2.0	7.0
Air.....	Over one row of ¼-in. tubes	.....	9	14	35
Air.....	Over one row of 1-in. tubes	. ...	5	8	19
Air.....	Over several rows of 1-in. tubes, with or without fins (staggered)	....	8	12	30
Water..	Through ¼-in. tube	750	1,500	2,600	
Water..	Through 1-in. pipe	575	1,150	2,000	
Water..	Through 3-in. pipe	450	900	1,500	
Water..	Over several rows of 1-in. tubes	1,000	1,200	2,400	

Part II. Natural Convection in Air (Convection *plus* radiation)

Shapes	Surface finish	Temperature difference		
		50°	100°	200°
Vertical walls of furnaces, tanks, etc.	Nonmetallic	1.6	2.0	2.6
Vertical walls of furnaces, tanks, etc.	Aluminum paint	1.2	1.4	1.7
Horizontal tubes, pipes, ducts, pipe covering, 1 in. OD..............	Nonmetallic	2.2	2.5	3.0
Horizontal tubes, pipes, ducts, pipe covering, 2 in. OD..............	Nonmetallic	2.1	2.3	2.8
Horizontal tubes, pipes, ducts, pipe covering, 6 in. OD..............	Nonmetallic	1.9	2.1	2.6
Horizontal tubes, pipes, ducts, pipe covering, 2 in. OD..............	Aluminum paint	1.6	1.8	2.1

Building walls: Inside, $h = 1.65$; outside, $h = 6.0$

Part III. Condensation and Evaporation (Usual velocities and pipe sizes)

Condensing steam, air and gas free....	2,500
Condensing or evaporating steam (ordinary amounts of entrained air).................................	1,000 to 2,000
Condensing or evaporating ammonia...................	300 to 600
Condensing or evaporating freon.......................	150 to 400

Caution.—Table XVII gives different kinds of surface coefficients. Those given for forced convection are true film conductances for convection only, whereas *the coefficients for natural convection in air include radiation.*

The true convection coefficients in Part I of Table XVII represent a very brief summary of the hundreds of pages of experimental data on such coefficients. Many variables affect film conductance, including fluid velocity, viscosity, conductivity, specific heat, density, etc. Correlations of experimental data are made on the basis of dimensionless criteria such as the Reynolds number, $DV\rho/\mu$ (page 259), the Prandtl number $C\mu/k$, the Nusselt number hD/k, and others. Thus, the curve of Fig. 129 (logarithmic coordinates) gives with fair accuracy the equation of forced-convection heat transfer for air, water, superheated steam, oil, and other fluids flowing in pipes of any diameter and at any velocity above the critical value at which viscous flow occurs.

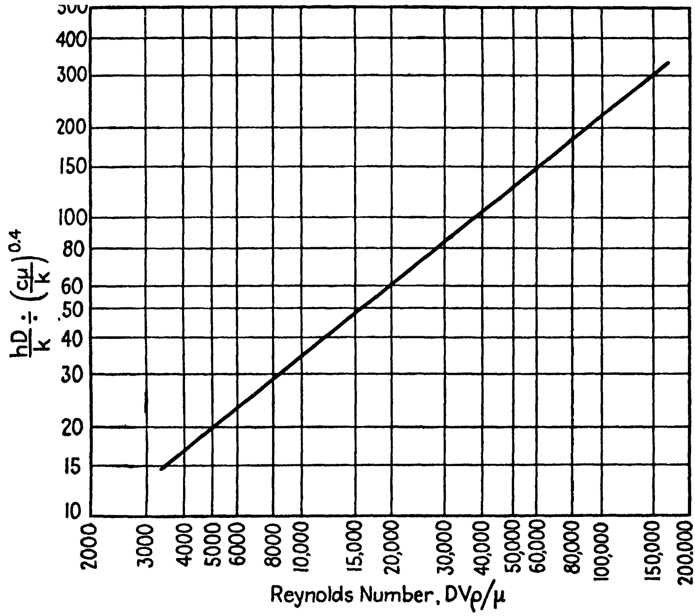

FIG. 129.—Convection coefficient h for fluids in pipes. (Applies to any liquid, gas, or vapor flowing in any pipe 0.5 to 4.0 in. diameter.)

The coefficients of Table XVII, Part II, are limited to the cases of heat transfer to or from a surface in "still air." Both convection and radiation are included, since in most practical problems the total heat transfer is required.

IV. SOLUTION OF PRACTICAL ENGINEERING PROBLEMS

The two common types of engineering problems deal with: (1) **insulation,** or *minimizing heat transfer;* and (2) **heat exchangers,** in which the object is to secure the *maximum heat transfer.* In either case the heat-flow path will probably consist of several parts, with heat flowing through them in series or in parallel. The analysis is again similar to that of electric circuits, using the following principles:

1. *For a* **series** *heat-flow path,* **resistances** *are additive:*

$$R_T = R_1 + R_2 + R_3 \cdots \text{etc.}$$

Or, since conductance is the reciprocal of resistance:

$$\frac{1}{C_T} = \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} \cdots \text{etc.}$$

$$C_T = \frac{1}{\frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} \cdots \text{etc.}} \tag{29}$$

Such an equation may be used to find the over-all conductance of any number of individual resistances (or conductances) in series. For a solid of conductivity k, the individual conductance is: $C = kA/L$, where A is the area and L is the thickness or length of path.

2. *For* **parallel** *heat-flow paths,* **conductances** *are additive:*

$$\begin{aligned} C_T &= C_1 + C_2 + C_3 \cdots \text{etc.} \\ &= \frac{k_1A_1}{L_1} + \frac{k_2A_2}{L_2} + \frac{k_3A_3}{L_3} \cdots \text{etc.} \end{aligned} \tag{30}$$

In calculating the *conductance per unit area,* the individual areas are decimals (percentages). Equations (29) and (30) may be combined for calculating the over-all conductance of assemblies such as a wood-frame building wall, or a hollow-tile wall, in which there are both series and parallel heat-flow paths (see example, following).

The over-all conductance per unit area of a wall is usually called the **over-all coefficient or thermal transmittance,** symbol

U. For a single homogeneous wall separating two fluids, the over-all coefficient is composed of two surface coefficients, h_1 and h_2, and the conductance of the solid section which has a conductivity k, and a thickness L (the area A being unity). Then

$$U = \frac{1}{\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}} \tag{31}$$

and

$$q = UA(T_1 - T_2). \tag{32}$$

U is therefore in Btu per square foot per hour per degree mean temperature difference between the two fluids. This equation for the over-all coefficient may be expanded to include any number of conductances, and it is, of course, a form of the series-flow equation (29). Table XVIII gives values of U for some of the common cases of engineering practice.

Example of Heat-flow Calculations.—Compare the heat loss of the following walls in Btu per hour per 100 sq ft of wall: (*a*) solid concrete wall 10 in. thick; (*b*) wood-frame wall with ¾-in. plaster inside, 2 × 4 pine studs on 16-in. centers, 1-in. matched pine siding outside, and stud spaces filled with mineral wool. Assume 70° inside and 0°F outside the wall in both cases.

Solution.—*a*. From Table XVI, the conductivity of concrete is $k = 12$. From Table XVII, the surface coefficients for a building wall are: Inside, $h_1 = 1.65$; outside, $h_2 = 6.0$. Then the *over-all coefficient* of the wall is

$$U = \frac{1}{\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}}$$

$$= \frac{1}{\frac{1}{1.65} + \frac{10}{12} + \frac{1}{6.0}} = \frac{1}{0.606 + 0.833 + 0.167} = \frac{1}{1.606} = 0.624$$

$$q = UA(T_1 - T_2) = 0.624 \times 100(70 - 0) = 4368 \text{ Btu per hr per 100 sq ft}$$

b. The surface coefficients are, as before, $h_1 = 1.65$ and $h_2 = 6$ (Table XVII). From Table XVI the conductivity of pine wood is $k = 0.8$, that of plaster is $k = 3.3$ and that of mineral wool fill is $k = 0.27$. A 2 × 4 actually measures 1.625 × 3.625 in., and a 1-in. board measures 0.78 in. thick. The conductance of the ¾-in. plaster is then

$$C_p = \frac{k}{L} = \frac{3.3}{0.75} = 4.4.$$

TABLE XVIII.—OVER-ALL COEFFICIENTS OR "TRANSMITTANCES"

(U = Btu transmitted through wall per hour per square foot per degree Fahrenheit temperature difference between the two fluids)

	Coefficient U
Furnace Walls:	
9-in. firebrick, 4-in. common brick	0.4
13½-in. firebrick, 4-in. common brick	0.3
9-in. firebrick, 4½-in. insulating brick, 4-in. common brick	0.11
Bare and Covered Pipes (1½ to 3 in. nominal diameter. Coefficient based on external pipe area):	
Bare black pipe: 2 lb sat steam	2.4
Bare black pipe: 100 lb sat steam	3.0
Bare black pipe: 250 lb sat steam	3.5
Asbestos air cell, 4 ply, 1 in. thick	0.65
Laminated asbestos, 1 in. thick	0.5
85 per cent magnesia, 1 in. thick	0.45
85 per cent magnesia, 2 in. thick	0.3
Building Walls (15 mph wind):	
8-in. brick, ½-in. plaster	0.46
8-in. brick, furred and plastered	0.30
12-in. brick, ½-in. plaster	0.34
8-in. stone, ½-in. plaster	0.64
10-in. concrete, ½-in. plaster	0.57
Ordinary frame construction (dwelling), siding or brick veneer	0.26
Same frame construction with ½-in. insulation	0.17
Same frame construction with Al foil facing stud space	0 19
Same frame construction with 4-in. mineral wool	0 07
Stucco over 8-in. hollow tile, ½-in. plaster	0 38
Corrugated steel (wall or roof)	1.5
Single window glass	1.13
Double windows	0.45
Hollow glass tile, 6 × 6 × 12 in. thick	0.60
Building Roofs:	
Shingle roof on 1-in. boards, plastered ceiling	0.28
Same roof and ceiling with 2-in. insulation between	0.10
Composition roof on 2-in. cement tile	0.84
Composition roof on 4-in. concrete	0.64
Composition roof on 1-in. boards	0.49

Heat Exchangers:	Velocity of water, fps	
Steam surface condensers and evaporators	4	500
	8	700
Ammonia condensers and evaporators	4	225
	8	300
Freon condensers and evaporators	4	250
	8	275

The conductance of the wood siding is $C_s = k/L = 0.8/0.78 = 1.023$.

The studs and the mineral wool fill constitute a parallel path consisting of 1.625/16 or 10.2 per cent pine wood and 14.375/16 or 89.8 per cent mineral wool. The *unit conductance* of this mid-section, consisting of studs plus mineral wool, by Eq. (29) is

$$\begin{aligned} C_m &= C_1 + C_2 = \frac{(k_1A_1)}{(L_1)} + \frac{(k_2A_2)}{(L_2)}, \\ &= \frac{(0.80 \times 0.102)}{3.625} + \frac{(0.27 \times 0.898)}{3.625}, \\ &= 0.0225 + 0.0668 = 0.089 \text{ Btu per sq ft per hr per sq ft of wall area.} \end{aligned}$$

The *over-all coefficient* for the wall is

$$\begin{aligned} U &= \frac{1}{\dfrac{1}{h_1} + \dfrac{1}{C_p} + \dfrac{1}{C_m} + \dfrac{1}{C_s} + \dfrac{1}{h_2}} \\ &= \frac{1}{\dfrac{1}{1.65} + \dfrac{1}{4.4} + \dfrac{1}{0.089} + \dfrac{1}{1.023} + \dfrac{1}{6.0}} \\ &= \frac{1}{0.606 + 0.227 + 11.23 + 0.97 + 0.167} \\ &= 1/13.20 = 0.0757 \text{ Btu per sq ft per hr per deg.} \\ q &= UA(T_1 - T_2) = 0.0757 \times 100(70 - 0) = 530 \text{ Btu per hr.} \end{aligned}$$

Hence the 10-in. concrete wall will lose 4368/530 = 8.24 times as much heat as the insulated frame construction.

The **temperature difference in heat exchangers** is not uniform as it usually is in the case of a building wall. Hence in Eq. (32) a *mean temperature difference* must be used instead of the simple difference $T_1 - T_2$. Take as a typical example a boiler plant *economizer*, in which water on its way to a steam boiler is heated by the flue gases leaving the boiler setting. It is evident that the water may flow either in the same direction as the gases or in the opposite or counter direction. These arrangements are known as *parallel flow* and *counterflow*. In either case the water is becoming progressively hotter and the gases are becoming cooler, in the direction of their flow. At every point in the exchanger, there is a certain temperature difference between water and gases. But the only places where these temperature differences may be conveniently measured are at the inlet and the outlet, *i.e.*, at the two extreme ends of the unit. Let us call the larger temperature difference (at one end) ΔT_1 and the lesser temperature difference (at the other end) ΔT_2. There *are two ways of computing the average or mean temperature difference:*

1. Arithmetical mean temperature difference:

$$\text{Arith mtd} = \frac{\Delta T_1 + \Delta T_2}{2}. \tag{33}$$

2. Logarithmic mean temperature difference:

$$\text{Log mtd} = \frac{\Delta T_1 - \Delta T_2}{\log_e \dfrac{\Delta T_1}{\Delta T_2}}. \tag{34}$$

For applications of the logarithmic mean see Experiment 48, page 212, and Experiment 81, page 418.

EXPERIMENT 48. Heat Losses from Bare and Covered Pipes

I. OBJECT

The object of this experiment is to compare the insulating value of various steam-pipe coverings by determining the over-all coefficient of heat transmission of each covering and the unit conductivity for each insulating material.

II. PREFACE

Commercial pipe coverings differ widely as to the material, character of construction, and method of fabrication. A good pipe covering, in addition to being a good insulator, should be fireproof, waterproof, vermin proof, odorless, and light in weight. It should also be mechanically strong and should suffer no loss of insulating value with age. Asbestos and carbonate of magnesia are the most commonly used pipe-covering materials, but there are also a number of others, among which might be mentioned hair felt, cork, mineral wool, wood, and paper.

The only logical method for testing commercial pipe coverings is of course to mount these coverings on pipe of the size for which they were intended. Extensive tests of commercial coverings have been made by various investigators, and two general methods for heat measurement have been used. For steam-pipe coverings, the most natural method would be to fill the covered pipe with steam, to measure the heat content of the steam entering and leaving the test section, and to condense and weigh the steam. For economical reasons a dead-end pipe is usually used, the test pipe itself acting as the steam condenser. The second and more accurate method is to supply and measure the heat electrically. When the latter method is used, the "end correction" is eliminated by covering the pipe ends with heavily insulated caps and maintaining separately heated end sections adjacent to the caps at the same temperature as the test section. The heat from the main test section cannot then travel along the pipe and must escape radially through the covering under test.

In the present experiment the steam-condensation method will be used because of the simplicity of the apparatus involved and because this apparatus more fittingly illustrates the use of pipe coverings in practice.

III. ANALYSIS

The problem of heat flow from a covered steam pipe is a simple case of resistances in series. Applying Eq. (29) and examining each resistance separately (as represented by each term in the denominator), the following is a brief summary:

1. **Surface Resistance between Steam and Inside Wall of Pipe.**—For wet steam this resistance is very low on account of the relatively high conductivity of the water film, and consequently the temperature drop is negligible. With superheated steam the resistance is increased, but it is still so low that it is usually neglected in dealing with covered pipes. In the case of bare pipes, this resistance is of greater importance. The temperature difference between the steam and the inside wall of the pipe will not be determined in this experiment on account of experimental difficulties. The value for the inside-surface conductance may instead be taken directly from Table XVII, page 204 (for condensing steam).

2. **Conductor Resistance of the Metal in the Pipe Wall.**—Here again the low resistance of the pipe-wall material introduces experimental difficulties in the measurement of the difference between the inside and outside pipe-surface temperatures. The value of the conductivity for the steel pipe wall will therefore be taken directly from Table XVI, page 202.

3. **Contact Resistance between the Pipe and Covering.**—This resistance will of course depend on how closely the covering fits the pipe. For small air spaces, such as occur with well-fitted commercial coverings, it has been found that the contact resistance plus the resistance of the air space is approximately equal to the resistance of an equivalent additional thickness of covering. In other words, the total resistance is the same as though there were no air space and the temperature of the inside surface of the covering were the same as the temperature of the pipe. This assumption will be used in the present experiment, and the thickness of covering will be taken as half the difference between the outside diameters of covering and pipe. If a large air space (say ¼ in. or more) is left between pipe and covering, this space has little value as an insulator. It has been demon-

strated experimentally that a loose-fitting covering actually results in a greater heat loss than a tight-fitting covering of the same thickness, even though the former contains more insulating material. This is due to convection and to the fact that the area of the heat path through the conductor (*i.e.*, the covering) has been increased while its length remained the same.

4. Insulation Resistance of the Material Used for Covering the Pipe.—In attempting to evaluate this term, it is at once seen that the area of the heat path increases from the inner to the outer surface of the covering. It will be necessary, then, to use an average or mean value of the area. An analysis by use of the calculus will show that the **logarithmic mean area** should be used, *i.e.*:

$$A_{lm} = \frac{A_2 - A_1}{\log_e \dfrac{A_2}{A_1}}. \tag{35}$$

The logarithmic mean is frequently used in problems of heat transfer (see experiment on Surface Condenser Test, page 418), but examination will show that the arithmetical mean area,

$$A_{am} = \frac{A_1 + A_2}{2}, \tag{36}$$

gives a value checking the logarithmic mean within 4 per cent when A_2/A_1 is 2 or less. The arithmetical average is therefore sufficiently accurate, unless the pipe covering is very thick. The length of path L_d is equal to half the difference between outside diameters of covering and pipe, as explained above. The value of C_d, the conductivity of the insulating material, is in this case the unknown to be determined by experiment. The conductivity increases with temperature, and the exact value will depend on the temperature range used in the experiment.

5. Surface Resistance between Outside of Covering and Surrounding Air.—This is to be determined by experiment and compared with the corresponding values taken from Table XVII, page 204.

IV. PREPARATIONS

Apparatus.—Figure 130 shows the general arrangement of one unit of the steam-condensation apparatus. As many units are used as there are pipe coverings to be tested simultaneously, plus one unit operating without covering. The other apparatus required are as follows: thermometers, thermocouple pyrometer, separating steam calorimeter, steel tape, steel scale, and outside calipers.

V. INSTRUCTIONS

1. Procedure.—With the drain valve open wide, open steam valve a small amount and allow steam to blow through long enough to purge apparatus of all air; then close drain valve. Place thermometers to obtain air temperature. Connect thermocouple pyrometer for observing temperature of the steam, temperature of outer surface of the pipe, and that of outer surface of the covering. When conditions have become uniform, as shown by constant temperature readings, the condensate level in the gage glass is noted and the run started.

Complete measurements of the pipe and covering should be taken and recorded. Be sure to obtain also all other data called for on Form 48. Calibration of condensate chambers should be obtained, in terms of weight of condensate per inch of water in the gage glass.

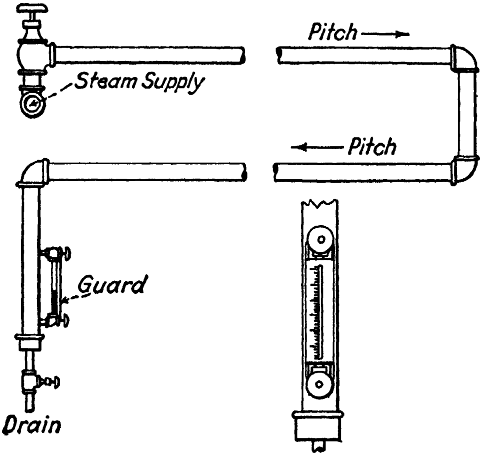

FIG. 130.—Steam-condensation apparatus for pipe-covering tests.

2. Determinations.—Duplicate determinations of at least 1 hr each should be made on each covering and on the bare pipe. All units may be run simultaneously over a 2-hr period with readings every 10 min. On account of the large number of readings, much care is necessary in arranging and recording the data.

3. Results and Calculations.—For a suggested tabular arrangement of data and results, see Form 48. If the condensate chamber is made from the same size pipe as the test section and is covered with the same material, it may be assumed that the test section extends to the average water level (make measurements accordingly), and that the condensate leaves the test section at steam temperature. *The end correction will then be zero.* Any cooling of the water below steam temperature is due to heat loss through the walls of the condensate chamber below the water level and should not be charged to the test section. The top surface of the condensate will be very close to steam temperature, and the heat loss from steam to condensate at this surface may be neglected. If the test section is comparatively long (say at least 25 ft over all), the percentage of error due to these assumptions will be negligible, as they very closely represent the actual operation.

The areas of the elbows in the test section are to be computed as closely as possible and added to the area of the pipe.

The efficiency of the insulation is defined as follows:

$$E = \frac{(\text{heat lost from bare pipe}) - (\text{heat lost from covered pipe})}{(\text{heat lost from bare pipe})}$$
$$= \frac{\text{heat saved by insulation}}{\text{heat lost without insulation}}. \qquad (37)$$

The heat-transfer coefficients to be calculated for each test pipe are: (1) the over-all coefficient U [Eqs. (31) and (32)], (2) the conductivity of the insulating material k [Eq. (26)], and (3) the outside-surface coefficient h [Eq. (28)].

For the calculation of the conductivity of the insulating material, the inside (pipe side) surface temperature of the material is needed. If this was not measured, it may be closely approximated as follows: Using the film coefficient for condensing steam from Table XVII, page 204, the total heat transmission (item 19, Form 48), the steam temperature (item 11), and the inside area of the pipe, solve for the inside-surface temperature of the pipe by Eq. (28), page 203. Using this result, select a value for the conductivity of the steel pipe from Table XVI, page 202, and calculate the outside-surface temperature of the pipe by Eq. (26), page 200. Consider items 12 and 13 identical, as already suggested, and solve Eq. (26), for the conductivity of the covering (see page 202).

When these calculations have been completed, they should be checked by recalculating the over-all coefficients from the individual coefficients by Eq. (31), page 207.

The report should include a temperature gradient sketch (see Fig. 126, page 195) for the test pipe covered with 85 per cent magnesia.

Form 48.—Condensation Test of Steam-pipe Covering

Test performed by.......... ....Barometer.............Hg.
Nominal pipe size..........Equivalent length of test section ..... ft.

Item number	Item	Symbol	Units	Values	
1	Run number........				
2	Duration of run...		Min.		
3	Thickness of covering	t_2	In.		
4	Thickness of pipe...	t_1	In.		
5	Name of covering				
6	Inside area of pipe per linear ft	A_1	Sq. ft.		
7	Outside area pipe per linear ft .	A_2	Sq. ft.		
8	Outside area covering per linear ft........................	A_3	Sq. ft.		
9	Gage pressure of entering steam	..	Lb. per sq. in.		
10	Quality of entering steam.. ...	.	Per cent		
11	Saturation temperature of entering steam..................	T_1	Deg. fahr.		
12	Temperature of outside of pipe.	T_2	Deg. fahr.		
13	Temperature of inside of covering........................				
14	Temperature of outside of covering.......................	T_3	Deg. fahr.		
15	Room temperature.....	T_4	Deg. fahr.		
16	Temperature of condensate .	T_c	Deg. fahr.		
17	Wet steam condensed per hr ...	..	Lb.		
18	Dry steam condensed per hr...	..	Lb.		
19	Total heat transfer per hr.....	..	B.t.u.		
20	Heat transmission per sq. ft. of pipe surface per hr...........	q	B.t.u.		
21	Over-all coefficient of heat transmission, B.t.u. per sq. ft. of pipe surface per hr. per deg........	U			
22	Efficiency of insulation.........	E	Per cent		
23	Conductivity of insulating material, B.t.u. per sq. ft. per hr. per deg. per in. in thickness......	k			
24	Surface coefficient by test.....	h			

Date.................... Observers:..................................
....
....

VI. NOTES AND PRECAUTIONS

1. The **same amount of condensate** should be accumulated each successive 10 min by a given test pipe. If these amounts do not check after a reasonable warming-up period, look for the trouble.
2. The **most likely source of error** is insufficient venting of air before starting. See that each test pipe is blown down thoroughly.
3. Use several thermometers for air temperature, place them on a level with the test sections, but protect them from radiation. (A piece of aluminum foil makes a good shield for the thermometer bulb.)
4. Do not open doors or windows near the test unit during the test.
5. It is always difficult to obtain representative steam samples for the calorimeter. If each test unit is fed upward through a large pipe connection from the steam supply (Fig. 130), the quality of the entering steam should be very nearly 100 per cent.

EXPERIMENT 49. Test of a Direct Steam Radiator

I. OBJECT

The object of this experiment is to familiarize the student with some of the common facts about steam-radiator heating by actual test of a radiator under normal operating conditions.

II. PREFACE

Cast-iron radiators, of more or less standardized design and construction, are extensively used for the heating of buildings. The heating medium used in these radiators is either steam or hot water, and a variety of systems is used in obtaining circulation and convenient control of the heating medium.

Although the present test is concerned with a "direct" cast-iron radiator without an enclosure, the test methods and procedure are the same for the indirect or enclosed types. The present trend in radiator installation is toward the use of the recessed indirect or convector type, and the heating element in this case may be either a ferrous or a nonferrous unit.

Steam radiators are supplied with low-pressure steam. The steam pressure is seldom greater than 5 lb gage, and in "vapor" systems or "vacuum" systems it is frequently atmospheric or below.

In "two-pipe" steam-heating systems, the condensate returns to the boiler or is discharged through a "return" pipe connected to the opposite end of the radiator from the steam-supply line.

A **"trap"** is located between the radiator and the return line, and this trap allows water to enter the return but closes when steam reaches it. In a "one-pipe" system the condensate and the steam are carried in the same pipe, the condensate returning to the boiler against the steam flow.

Radiators are designated by the number of "columns" or "tubes" per "section," by the number of sections composing the radiator, and by the approximate height of the radiator from the floor line to the top (*e.g.*, "a 3-column, 20-section, 38-in. radiator"). For certain types of service, special radiators are preferred, such as wall and ceiling radiators, hospital radiators, pipe-coil radiators, and decorative narrow-column radiators.

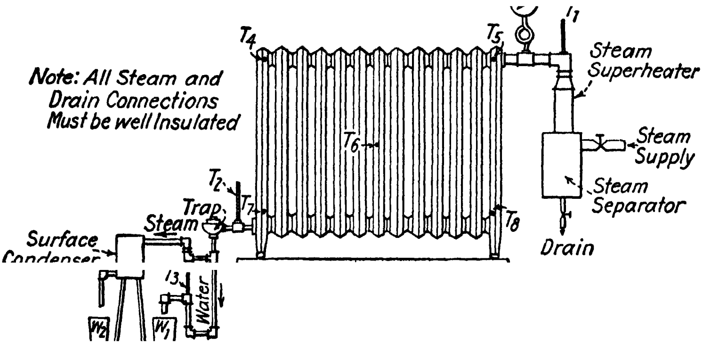

FIG. 131.—Setup for test of steam radiator.

The American Society of Heating and Ventilating Engineers has selected 240 Btu per hr as a standard value for the heat given off by 1 sq ft of radiator surface. The actual unit heat transfer varies rather widely, however, with the type and size of radiator, the temperature difference between the steam and the outside air, the radiator location, its surface finish, etc. (see Table XVII, page 204).

The A.S.H.V.E. publishes Standard Codes for the testing and rating of both direct and concealed radiators. These Codes define the heat-output rating in terms of square feet of "equivalent direct radiation" (E.D.R.) as the total Btu given off per hour divided by 240. Most radiator manufacturers give in their catalogues the capacity ratings for their units in square feet E.D.R. The A.S.H.V.E. has more recently adopted a new unit of capacity, the Mb, which is to replace the square foot E.D.R. One Mb is equal to 1000 Btu.

III. PREPARATIONS

Figure 131 shows a setup for this test. There is some choice regarding certain details of this setup. The trap and condenser may be replaced by a condensate receiver with gage glass and valve, or preferably by a plain U-tube water leg of sufficient height to resist the maximum pressure. An electric superheater, located between the steam separator and the inlet thermometer, is a decided improvement, since the steam calorimeter is not satisfactory on low pressures. Scales reading to 0.01 lb (or to ¼ oz) should be provided for weighing the condensate.

IV. INSTRUCTIONS

1. Procedure.—As indicated in Fig. 131 and on Form 49, readings should include pressure, temperature, and quality or superheat of entering steam, temperature in all parts of the radiator where thermometer wells are provided, temperature of condensate, weight of condensate, weight of steam passing trap (if any), room temperature, and barometric pressure. Be sure to secure all descriptive information called for on blank, including the actual surface areas.

Place apparatus in operation by opening condensate drain and calorimeter valves wide and cracking main steam and separator drain valves on supply line. When radiator has warmed up, set pressure-regulating valve for desired supply pressure and open main valve wide. Open vent valves and allow same to blow until radiator is purged of air. Hold all conditions constant for at least 10 min before starting test.

2. Determinations.—If automatic steam-pressure regulation is available, a ½-hr run at each of four steam pressures makes a good test. Longer runs may be necessary if a trap is used, because of its periodic discharge. If the test is to run at one steam pressure only, it may be possible to vary the temperature of the air surrounding the radiator or to demonstrate the effect of air circulation by the use of an electric fan. Tests with and without a radiator cover will show the effect of such devices. Take all readings every 5 min.

3. Results and Calculations.—The required results are indicated on the result sheet, and methods of calculation should be apparent from the sequence of items on this sheet. For item 16, use Eq. (32), page 207.

The report should include a sketch showing the setup, with all instruments in place.

In the discussion the over-all coefficient of heat transmission should be compared with those given in Table XVIII, page 208, and the total heat given off should be compared with the value of 240 Btu per sq ft quoted above. A graph showing the variation of the over-all coefficient with temperature difference should be presented, and the effect of air circulation on capacity should be discussed (see Table XVII, page 204). The equivalent direct radiation should be compared with the manufacturer's rating.

V. PRECAUTIONS

1. See that all thermometer wells are filled with light oil (just enough to cover the thermometer bulb).
2. Be sure that the separator on the steam-supply line is kept drained so that no water enters the radiator.
3. Observe gage glasses connected to radiator sections to make sure that trap is functioning properly.
4. Open air-vent valves occasionally to make sure that no air collects in the radiator.
5. The **index of test accuracy** is the duplication of condensate weights in successive 5-min intervals. If large variations occur, correct the trouble, or use long test runs.
6. To prevent evaporation loss from hot condensate, the condensate may be discharged into pails containing a weighed quantity of cold water.

VI. NOTES

The Standard Code prescribes the following test conditions which are different from those suggested for the present test:

1. The radiator shall be set up in a special test booth, 12 by 15 ft by 9 ft high, inside dimensions, or in a refrigerated test room of like size, and the temperature of the air surrounding the test room shall be closely controlled.
2. A steam by-pass pipe shall be installed around the radiator, so that the latter may be cut out of the system and a no-load test made to determine the heat losses from the piping connections.
3. The steam supply shall be at 15.7 lb abs, with a superheat of 2 to 5 deg.
4. The duration of test shall be at least 1 hr, with the condensate weights during successive 10-min intervals checking to ±5 per cent of the average for the hour.

Form 49.—Results of Steam-radiator Test

Test conducted by..
Type of radiator..............................Made by.............
Number of columns....Number of sections....Height....Finish......
Surface per section.....sq ft. Total surface.....Barometer.......Hg

Item number	Item	Units	Values	
1	Run number.........................			
2	Duration of run......................	Min.		
3	Steam-supply pressure, gage..........	Lb. per sq. in.		
4	Steam quality, or superheat..........	% or deg.		
5	Radiator temperature, average........	Deg. fahr.		
6	Room temperature, average..........	Deg. fahr.		
7	Temperature difference...............	Deg.		
8	Weight of condensate................	Lb. per hr.		
9	Weight of steam passing trap.........	Lb. per hr.		
10	Trap efficiency......................	Per cent		
11	Temperature of condensate leaving trap	Deg. fahr.		
12	Heat extracted from 1 lb. of steam supplied to radiator....................	B.t.u.		
13	Total steam supplied................	Lb. per hr.		
14	Total heat given off by radiator per hr.	B.t.u.		
15	Heat transmission per sq. ft. per hr.. .	B.t.u.		
16	Over-all coefficient of heat transmission B.t.u. per sq. ft. per hr. per deg. temperature difference between steam and air........................			
17	Equivalent direct radiation....	Sq. ft.		
18	Radiator capacity...........	Mb		

Date................ Observers:..
....
....

5. The radiator shall be vented continuously.

6. The test capacity shall be corrected to standard conditions by assuming that the heat transfer varies as the 1.3 power of the temperature difference. (Standard temperature difference 145 deg.)

Although the radiator test is a simple one, extreme care is necessary if a high degree of accuracy and reproducible results are desired. Some of the common sources of error are: entrained water in the entering steam, insufficient venting, loss of condensate due to evaporation (flashing), excessive air currents around the radiator, and poor insulation of the pipe connections. Another difficulty is that of obtaining a steady flow of the condensate. If a steady steam pressure is available, an open U-tube condensate-discharge leg is the most accurate, and it also has the advantage of requiring no manual control. The Code calls for a closed receiver with gage glass and hand valve, but this specification of the Code might be questioned.

A very satisfactory electric inlet steam superheater may easily be made by winding resistance wire around a piece of 1¼- or 1½-in. pipe about 1 ft long. The pipe should first be covered with a thin layer of mica insulation, and the completed superheater covered with high-temperature insulation (asbestos or magnesia pipe covering may be used). The inside of the pipe should be tightly packed with lengths of straight copper tubing in order to provide ample superheating surface. For a 110-volt supply, about 25 ft of 24-gage nichrome wire in series with a slide-wire rheostat makes a satisfactory electrical system. By varying the slide wire, the steam superheat may be controlled to an exact degree. An electric immersion heater within the pipe is even better than an external heater, since it responds more quickly to adjustments. Fins or other extended surface should be added to a plain immersion heater. Another very satisfactory method of superheating the steam is to use an ordinary domestic gas water-heating unit for a superheater.

The importance of heat transfer by radiation, and hence of surface finish, may be demonstrated by preparing two sheet-metal slip covers to fit closely over the radiator, one cover being bright-tinned finish and the other painted with oil paint. The difference in heating capacity with one cover as compared with the other is then due to the difference in surface emissivity (see Table XIV, page 198).

EXPERIMENT 50. Heat-transfer Test of a Fin-tube Coil

I. PREFACE

In the process of heating or cooling air (or any gas), there is usually a great inequality in heat-transfer coefficients. For instance, when air is being heated by a steam coil, the surface coefficient on the air side will probably be less than 10, while the steam-side coefficient may be 2,000 or even more. A similar inequality exists when the fluid inside the tubes is hot or cold water or when it is a vapor refrigerant such as ammonia or freon.

The most promising way to increase the heat transfer per foot of tube length (or per square foot of face area of the coil) is to increase the air-side surface area. If fins or other extended surfaces are firmly attached to the air side of the tubes and the air motion over these fin surfaces is maintained by a fan or blower, the capacity of the coil may be increased several times without increasing the amount of tubing used or the space occupied by the coil.

For the rapid heating or cooling of air and gases the extended-surface heat exchanger has therefore become the standard equipment. Applications include unit heaters, air conditioners, flue-gas economizers, refrigerant condensers, and automobile radiators. Air-cooled automotive and airplane engines are in much the same class. On account of the great variety of designs and of operating conditions, careful tests of extended-surface coils are necessary to determine their performance.

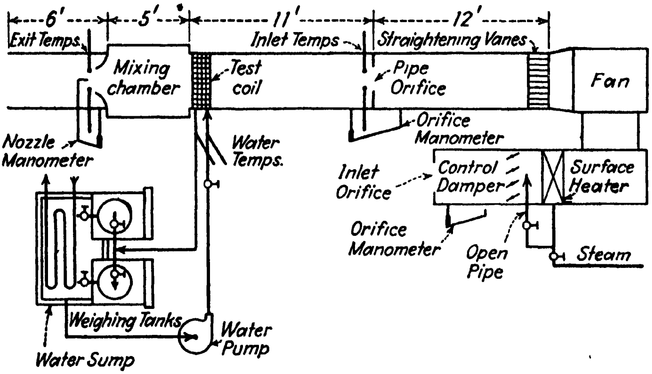

FIG. 132.—Diagram of setup for testing a duct-type fin-tube coil using water.

This experiment covers the testing of a typical unit heater, unit cooler, or separate air-heating or cooling coil, using water or condensing vapor (steam or refrigerant), as the heat carrier.

II. PREPARATIONS

1. Apparatus and Setup.—Figure 132 shows a typical setup for testing a duct-type coil, and Fig. 133 is from the A.S.H.V.E. Unit Heater Test Code. Essential instruments on the air side are dry-bulb and wet-bulb thermometers for air entering and air leaving the coil, a manometer for measuring pressure drop across the coil, and an air-metering orifice or nozzle with connected gages. A pitot-tube traverse or an anemometer traverse may be used for the air measurement if conditions permit. To check possible stratification and to obtain a better average, it is well to arrange four thermometers in the duct to read each of the air temperatures required. For testing unit heaters or coolers no inlet duct is necessary, but an auxiliary fan on the discharge duct or box is required if the full capacity of the unit is to be obtained (Fig. 133).

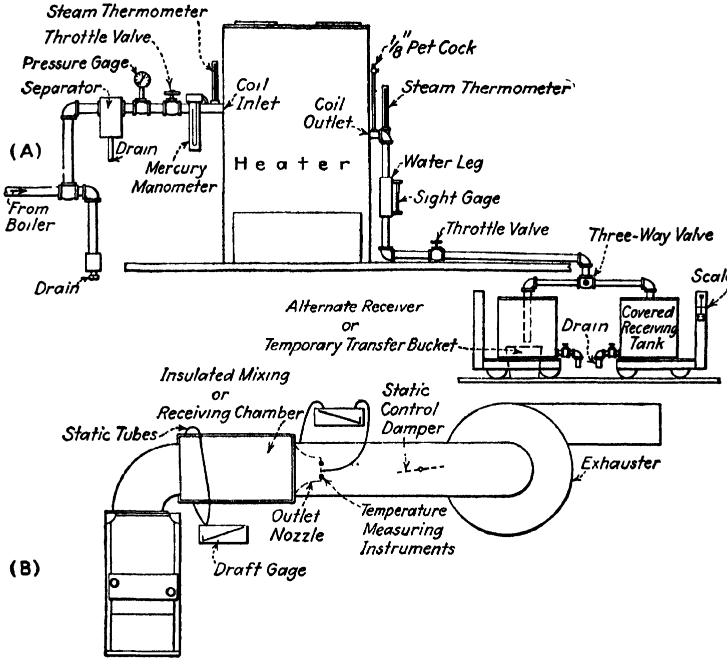

FIG. 133.—Arrangement of apparatus for unit heater test.

Measurements relating to the fluid inside the tubes will be as follows:

a. For water coils: Temperatures entering and leaving, differential pressure, weight of water in pounds per minute.

b. For steam coils: Pressure and temperature of steam entering (slightly superheated), and weight of condensate; also temperature of condensate.

c. For direct-expansion coils: Pressure and temperature of liquid entering expansion valve, temperature of mixture leaving expansion valve, and pressure and temperature of superheated vapor leaving coil. The quantity of refrigerant is difficult to measure, but may sometimes be approximated.

III. INSTRUCTIONS

1. Procedure.—The essential result of the test is the rate of heat transfer per square foot of air-side surface (or per square foot of face area) per degree mean temperature difference between the two fluids. For unit heaters and coolers, the capacity in Btu per hour under fixed operating conditions may be sufficient. Readings should be started as soon as possible and continued until data have been obtained for 1 hr of equilibrium operation (at each test condition) with readings every 5 min. For a series of tests, the independent variable is usually air velocity. Water velocity is an important variable for water coils. Temperatures may be varied if desired, to demonstrate that the heat-transfer coefficient is practically independent of temperature.

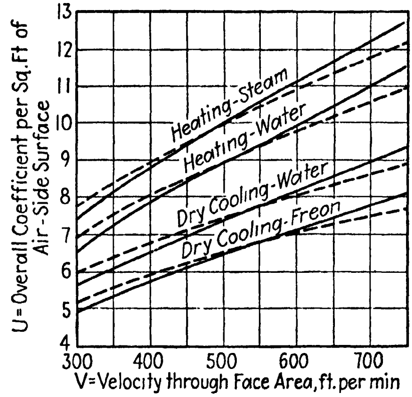

FIG. 134.—Typical heat-transfer coefficients for fin-tube air-heating and cooling coils. (Use dotted lines for coils with one or two rows of tubes.)

2. Results and Calculations.—Form 50 is a suggested summary of data and results. Heat-transfer coefficients should be calculated on the basis of logarithmic mean temperature difference [Eq. (34)]. The heat-transfer coefficient U should be plotted against the air velocity V on logarithmic coordinates. Based on the equation $U = CV^n$, the graph should be a straight line, and the slope n will usually be between 0.5 and 0.7. Typical coefficients for round-fin coils with thick fins are shown in Fig. 134. Coefficients for continuous-fin units on the same basis of air-side surface may be as much as 30 per cent lower. Rate of air flow as metered should check the air flow as calculated from the heat balance within 2 to 5 per cent. Heat balance by dry-bulb temperatures should check with 2 per cent of that calculated from wet-bulb readings (if there is no dehumidification).

Form 50.—Results of Fin-coil Test

Name of Coil ....Test No.........Date...... ..
No. rows of tubes..... .. ...Tube size........... Spacing............in. centers
Total air-side surface . . . .. sq ft. Gross face area... .. ..sq ft
Free area.........sq ft Primary tube surface..... sq ft Barom . . in. Hg

Average test conditions:

1	Dry bulb, °F	Air in...	. . Air out....
2	Wet bulb, °F	Air in .	..Air out .. . ..
3	Dew point, °F	Air in. . .	Air out . . . ..
4	Relative humidity, %	Air in..	. . Air out.. . ...
5	Sp. humidity, grains per lb dry air	Air in.	Air out ...
6	Enthalpy per lb dry air	Air in...	. Air out ....
7	Water temp: In	.Out.	. Mean.. ....

Calculated results:

8	Log mean temperature difference	. . . .
9	Weight of air mixture by orifice, lb per hr	. ...
10	Weight of dry air by orifice, lb per hr	. . . .
11	Weight of water, lb per hr	. . . .
12	Btu per hour, water basis	. .. .
13	Btu per hour, air basis, dry bulb	. .. .
14	Btu per hour, air basis, wet bulb	. . ..
15	Over-all heat transfer coefficient, *U*	.. ..
16	Face velocity, ft per min, average	. .. .
17	Air volume at standard density, cfm	.. ..

IV. NOTES AND PRECAUTIONS

1. All thermometer readings should be made to the nearest 0.1°F if possible.

2. The most likely source of error in fin-coil tests is due to stratification of exit air. Four thermometers at the discharge of the mixing box or exit duct should check within 1°F. The A.S.H.V.E. Test Codes specify a mixing box (20 to 90 air changes per minute), but the design and placing of baffles is a trial-and-error process. A disk-fan mixer in a round duct is quite satisfactory.

3. The entire duct section between the two sets of air thermometers should be insulated with 2 in. of hair felt or equivalent.

4. The diameter of the air-metering orifice should preferably be 60 to 75 per cent of the duct diameter, and the orifice should be preceded by 5 or more diameters of straight duct, with straightening vanes.

5. This experiment is not intended to cover the testing of dehumidifying coils, but the procedure in the two cases is almost identical. In a dehumidifying test an additional check on the latent-heat quantities is made possible by collecting and weighing the drip condensate.

6. For steam unit heaters the capacity is calculated on the basis of the latent heat of the steam only, no credit being given for desuperheating or for cooling of the liquid.
7. Air inlet and outlet conditions should not vary over 1° in 10 min, after the test has started.

EXPERIMENT 51. Test of a Steam Heating Boiler

I. PREFACE

Three of the important elements in a steam heating system have been considered in Experiments 48 and 49, pages 210 to 220), *i.e.*, the piping system, the natural-convection heaters or "radiators," and the forced-convection or unit heaters. The controls were also briefly classified in Table X, page 151. The other main unit in the system, the boiler, is also a heat-transfer unit, but its operation involves an additional process, that of combustion. Therefore a boiler test involves the usual measurements of fluid temperatures, pressures and quantities, plus the measurements necessary to determine the efficiency of combustion.

Heating boiler tests are covered by the Test Codes of the A.S.H.V.E., but since the method of testing is essentially the same as that for a power boiler, as described in Experiment 78, page 379, references will be made to the latter test, with only sufficient discussion to indicate the special requirements of the A.S.H.V.E. Codes.

II. PREPARATIONS

The boiler is to be set up according to manufacturers' directions, connected to a chimney of dimensions nearest those catalogued by the manufacturer (or a source of equivalent draft), and insulated 1½ in. thick as it normally would be in practice. Heating surfaces shall be cleaned before test. Figure 135, taken from the Code, illustrates the general setup, and the duties of the observers are essentially the same as those indicated in the boiler-test assignment schedule in Experiment 78, page 381, though fewer observers may be used in testing a small boiler.

III. INSTRUCTIONS

1. Procedure.—Two methods of firing are indicated in the Code, the "new-fire" method and the "continuous-firing" method. In most cases the continuous method is to be preferred, and the procedure is essentially as outlined under Procedure and in the Notes in Experiment 78, pages 382 and 391. The test is started with a "clean" fire and clean ashpit and with the least amount of fuel on the grate. The Code calls for a 2-lb operating pressure, and the weight of fuel charged at each firing is to conform to manufacturers' instructions.

2. Duration of Test.—For solid fuel the A.S.H.V.E. Code 1, covering the heat-balance test, requires a duration such that at least 140 lb of fuel has been fired for each square foot of grate area. For complete performance analysis, according to Code 3, five tests are required, one at "drive output" and the other four distributed over the operating range, with a total of 250 lb of coal fired per square foot of grate, for each test (or until the error in fuel and water determination shall not exceed 2 per cent).

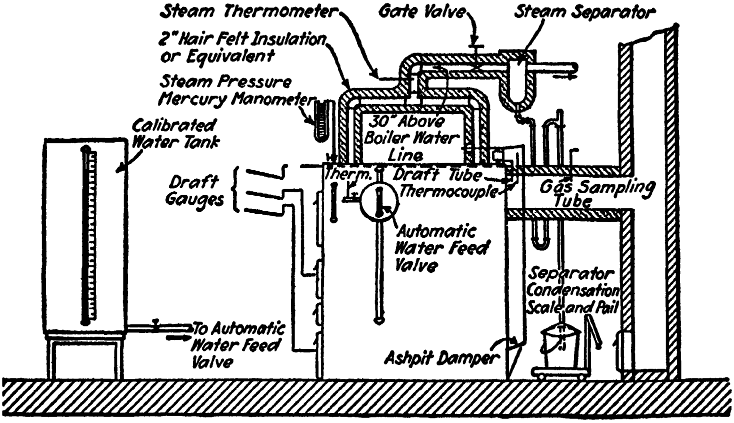
Steam Thermometer
Gate Valve
Steam Separator
2" Hair Felt Insulation or Equivalent
Steam Pressure Mercury Manometer
Calibrated Water Tank
30" Above Boiler Water Line
Draft Gauges
Therm.
Draft Tube
Thermocouple
Gas Sampling Tube
Automatic Water Feed Valve
Separator Condensation Scale and Pail
To Automatic Water Feed Valve
Ashpit Damper

FIG. 135.—Setup for test of heating boiler.

For tests with oil, the Code requires the burning of at least 2 lb of oil per cubic foot of furnace volume, and in no case a duration of less than 2 hr.

3. Results and Report.—The data and results called for by the Codes are almost identical with those listed in Forms 78*a* and *b*. The boiler description is of course slightly different, and performance is expressed in Btu per hour (or Mb) and in "square feet of steam radiation" instead of in boiler horsepower. The average interval between firings and the average weight of fuel per firing are also reported.

IV. NOTES

One method for testing solid-fuel heating boilers is to mount the entire boiler unit on platform scales, with all connections flexible. By this method a detailed analysis of performance is possible, and variations occurring between the firing periods may be accurately measured.

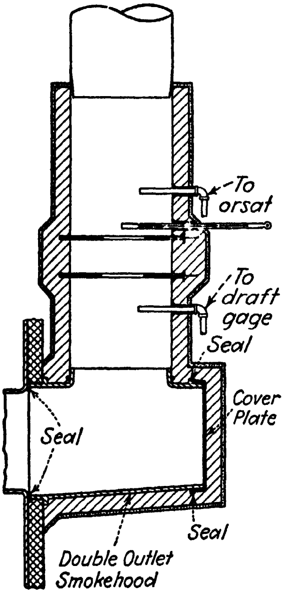

FIG. 136.—Insulated smoke hood with mixing orifices and instrument connections.

I.B.R. Rating Tests.—The Institute of Boiler and Radiator Manufacturers has adopted a Testing and Rating Code for low-pressure heating boilers. This Code covers all fuels, hand- and stoker-fired anthracite and bituminous coal and coke, also oil and gas. Following are some of the Code requirements:

Over-all efficiency shall not be less than 58 per cent for coal or coke fuel, hand fired, and not less than 68 per cent for oil. An oil burner shall be set to produce 10 per cent CO_2. If coal of less than 13,000 Btu per lb is to be used in the boiler, the net heat load must be multiplied by a correction factor before the rated boiler size is selected.

An insulating covering, 1.5 in. thick, is required over the entire smoke hood and stack connection during a test. For oil-fired units, two mixing orifices are required in the stack connection (each about $0.75D$), and the flue-gas temperature and gas analysis must be taken beyond these orifices, as shown in Fig. 136.

EXPERIMENT 52. Test of a Mechanical Warm-air Furnace

I. PREFACE

Warm-air heating, with mechanical air circulation by a centrifugal fan, has many advantages. The essentials of winter air conditioning, including positive heat distribution, ventilation, air filtering, humidification, and automatic temperature control may all be incorporated in a residence system at a very moderate cost.

The combustion chamber and heating elements of a mechanical warm-air unit may be made of cast iron or steel. Refractories are often used adjacent to the burners to protect the metal and to increase the heat transfer by introducing high-temperature radiation [compare Eqs. (25) and (28)]. Heat transfer by convection is aided by using fins or other extended surface, especially on the air side.

Double-wall insulating casings are used around the heating elements, to give protection from high temperatures. Fans, casings, plenum chambers, ducts, and registers should all be of adequate size for low resistance, hence giving a minimum of power consumption and noise. Installations should be made in accordance with the Codes of the National Warm Air Heating and Air Conditioning Association, which are the result of an impartial research program of long duration. Each installation is a special design problem, based on calculations of the heat losses from each room and involving a duct layout according to the house plans.

This experiment deals with the testing of a typical gas- or oil-fired warm-air unit with automatic control. Such units are rated according to their maximum fuel input in Btu per hour, and their fan capacity in cubic feet per minute. Automatic controls on these units usually include: (1) A bonnet thermostat that starts the blower at a temperature somewhere between 130 and 165° and stops the blower after the heating period, when the air temperature has again dropped to 135 to 100°. (2) A room thermostat that turns the burner on and off according to the heat requirements of the rooms. (3) A safety control that limits the maximum bonnet temperature to 200° or less, by shutting off the burners. (4) Ignition safety controls at the burners. Some of the newer installations are equipped with high-low-off controls instead of the simple on-off system, and the fan may be arranged for continuous operation to obtain more even heating and better ventilation.

In a laboratory test it is seldom practicable to have the unit connected to a complete duct system, but plenum chambers or stub ducts can be used and the resistance of the system simulated by using orifices on air intake and discharge.

II. PREPARATIONS

1. Apparatus.—Instruments required are Orsat, draft gages, thermometers, thermocouple pyrometer, revolution counter, fuel weighing or metering equipment, air orifices, anemometer, wattmeters.

2. Setup.—Connect Orsat apparatus to sampling tube at flue connection, below draft regulator (see Fig. 136). Place thermocouples for obtaining temperature of flue gases at sampling tube; also temperatures of combustion chamber, gas passages, etc. Place shielded thermometers or thermocouples for air temperatures. Connect wattmeters in motor circuits.

III. INSTRUCTIONS

1. Procedure.—Important results to be determined are burner capacity, fan capacity, over-all efficiency, and general heat balance. Tests should be made during equilibrium operation, with the controls set high or locked out. Air quantity and heated-air temperature are the most difficult measurements and special precautions should be taken to secure accuracy, using two

Form 52.—Results of Mechanical Warm-air Furnace Test

Manufactured by .... Fuel used ....
Manufacturer's rating: Btu per hr fuel input ....
Air volume .... cfm at .... in. static pressure

Operating Conditions:		
1	Fuel rate, gal per hr	
2	Fan speed, rpm	
3	CO_2 at flue connection, %	
4	CO at flue connection, %	
5	Draft at flue connection, in. H_2O	
6	Air-side static resistance, at bonnet (discharge), in. H_2O	
7	Air-side static resistance, at inlet above filters, in. H_2O	
8	Air-side static resistance across filters, in. H_2O	
9	Total air-side static resistance (6 + 7 + 8), in. H_2O	
10	Fuel pressure	
11	Flue-gas temperature at exit, °F	
12	Gas temperature leaving combustion chamber, °F	
13	Maximum combustion chamber temperature, °F	
14	Maximum refractory temperature, °F	
15	Maximum metal furnace wall temperature, °F	
16	Air-inlet temperature, °F	
17	Air-discharge temperature at bonnet, °F	
18	Air-temperature rise, °F	
Results:		
19	Pounds flue gases per lb fuel, actual (dry)	
20	Percentage excess air over theoretical	
21	Dry flue-gas loss, %	
22	Hydrogen and moisture losses, %	
23	Radiation and miscellaneous losses (plus unaccounted for)	
24	Percentage of heat to air = % efficiency	
25	Air volume handled by fan, cfm standard air	
26	Total electrical load, watts	

methods or instruments for each if possible. Flue-gas stratification is another source of error. For oil fuel, check the Orsat analyses by Fig. 192.

2. Determinations.—The independent variable in the tests may be fan speed, size of air orifice, burner capacity, or combustion adjustments. Ample time must be allowed for equilibrium for each run.

3. Results and Calculations.—Form 52 is a suggested tabulation of results. For combustion equations see Experiments 75 and 78.

IV. NOTES

The following questions are suggested as a basis for evaluation and discussion of results:

1. What is the capacity of the unit (*a*) In gallons of oil per hour? (*b*) In Btu per hour at bonnet? (*c*) In cfm of standard air?
2. Is the combustion complete? Is the air supply to the burner properly adjusted?
3. What are the losses, (*a*) To the dry flue gases? (*b*) Due to hydrogen and moisture? (*c*) From the casing? To the floor?
4. Are the operating temperatures satisfactory? Are metal or refractory temperatures dangerously high? (Not over 900° for metal or 1800° for refractory.) Is the air-outlet temperature too high or too low?
5. What would be the fuel cost and the electrical cost for 24 hr. steady operation?

CHAPTER VI

MEASUREMENT OF PROPERTIES OF GASES, VAPORS, AND AIR MIXTURES

PREFACE

The operation of many processes in industry is dependent upon the chemical and physical properties of gases and vapors. The engineer must be able to measure these properties if he is to deal quantitatively with the problems that arise in the fields of combustion of fuels, refrigeration, internal-combustion engines, air conditioning, drying, steam generation, welding of metals, etc. A variety of gases and vapors is involved, such as nitrogen, oxygen, carbon dioxide, carbon monoxide, ammonia, hydrogen, sulphur dioxide, methane, acetylene, water vapor, and mercury vapor. A mixture of gases, or of gas and vapor, often occurs, and hence the laws governing such mixtures must also be investigated by the engineer.

Most of the gas and vapor problems encountered in experimental mechanical engineering have to do either with the properties of a "working substance" in a heat-power machine or with the properties of atmospheric air or mixtures of atmospheric air with other gases and vapors.

Gases and Vapors as Working Fluids.—When gases and vapors are used in heat engines, the important properties are those that are associated with heat and work, *viz.*, pressure, volume, temperature, heat capacity, and, for wet vapors, quality. For the so-called "permanent gases" and highly superheated vapors, the relations of the properties are usually worked out by the laws of "perfect gases," and these relations are summarized in Table XX, page 235. Other constants for the common real gases are given in Table XIX. Problems dealing with the working fluids usually involve three processes: (1) flow of the fluid, as treated in Chap. VII, (2) flow of heat, as treated in Chap. V, (3) expansion or compression, as measured experimentally by the indicator diagram (Chap. II, pages 86 to 97).

Even for practical cases in which the laws of perfect or ideal gases do not apply exactly, the form of these laws is retained, and corrections for real gases and vapors are made by empirical factors. Thus the fluid-meter equations given in Table XXII, page 270 are all based on the ideal-gas laws, with coefficients and correction factors applied as indicated. Similarly the properties of real gases and vapors are determined experimentally, and these constants (given in Table XIX) are used in the ideal-gas equations. The common equation of state for an ideal gas is

TABLE XIX.—PROPERTIES OF GASES AND VAPORS

(Average values, room temperature)

	Chemical symbol	Molecular weight	c_p, Btu	c_v, Btu	$k = \frac{c_p}{c_v}$	$R = c_p - c_v$ ft-lb	Viscosity, centipoises
Air......	......	28.95	0.24	0.171	1.4	53.3	0.018
Acetylene	C_2H_2	26.02	0.38	0.30	1.25	59.	0.010
Ammonia..........	NH_3	17.03	0.52	0.40	1.3	91.	0.010
Butane	C_4H_{10}	58.08	0.35	0.32	1.3	26.	0.008
Carbon dioxide ...	CO_2	44.00	0.20	0.15	1.3	35.	0.015
Carbon monoxide..	CO	28.00	0.25	0.18	1.4	55.	0.018
Chlorine	Cl_2	70.91	0.12	0.09	1.33	22.	0.013
Ethane...	C_2H_6	30.05	0.39	0.32	1.3	51.	0.010
Ethylene	C_2H_4	28.03	0.40	0.33	1.25	55.	0.0104
Freon (12).	CCl_2F_2	120.9	0.15	0.13	1.14	13.	0.013
Helium.............	He	4.00	1.25	0.75	1.66	385.	0.020
Hydrogen..	H_2	2.01	3.4	2.4	1.4	765.	0.0089
Methane	CH_4	16.03	0.54	0.42	1.3	96.	0.011
Nitrogen....	N_2	28.02	0.245	0.175	1.4	55.	0.018
Oxygen......	O_2	32.00	0.215	0.155	1.4	48.	0.020
Sulphur dioxide....	SO_2	64.06	0.154	0.123	1.25	24	0.013
Steam (212°)...	H_2O	18.02	0.47	0.36	1.3	86.	0.011

For Other Gas and Vapor Data, see the following:

Steam tables, pages 473 to 479. **Viscosities, page 236.**
Air-vapor mixtures, pages 248 and 480. **Gas equations, page 235.**
Heat conductivities, page 202. **Flow equations, page 270.**

$$PV = wRT_1, \text{ or } \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}. \qquad (38)$$

The equation of path of a gas on the *PV* plane, representing the expansion line or the compression line on an ideal indicator card, is

$$PV^n = \text{a constant, or } P_1V_1^n = P_2V_2^n \qquad (39)$$

In these equations *P* is in pounds per square foot, *V* is in cubic feet, and w is in pounds. R is the difference in foot-pounds between the specific heat at constant pressure and the specific heat at constant volume, and is approximately equal to 1,544/molecular weight. R is practically independent of temperature, but c_p and c_v and their ratio $c_p/c_v = k$, all vary with temperature, as shown in Fig. 137.

For constant entropy adiabatic processes, as assumed in the flow-nozzle equations (page 269), and in the ideal internal-combustion engine cycles (pages 457 and 463), the exponent n in Eq. (39) becomes equal ratio of the specific heats, *i.e.*, $n = k$ (see also page 90).

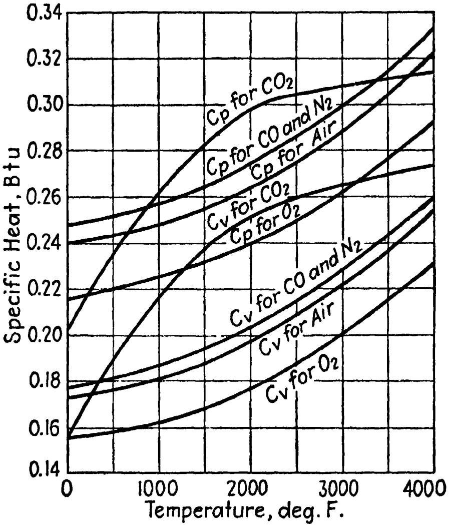

FIG. 137.—Specific heats of gases.

Other forms of the ideal-gas equations are given in Table XX.

Properties of gases and vapors at atmospheric pressure and room temperature are given in Table XIX. Variations of specific heat with temperature are given in Fig. 137, and variations of viscosity in the range to 220°F are given in Fig. 138.

For saturated vapors it is necessary to abandon the gas laws as a means of calculation and also to introduce an additional property, that of quality. Pressure and temperature are no longer independent properties, and this dependence of pressure and temperature is illustrated in Experiment 53.

TABLE XX.—IDEAL-GAS PROCESSES AND EQUATIONS

Gas processes	Equations	Final pressure	Final volume	Final temperature	Heat added, B.t.u.	External work, ft-lb.
Isothermal: $T = C$ $n = 1$	$PV = C$	$P_2 = P_1\frac{V_1}{V_2}$ Decreases	$V_2 = V_1\frac{P_1}{P_2}$ Increases	$T_2 = T_1$ No change	$\frac{RT_1}{J}\log_e\frac{V_2}{V_1}$ or $\frac{P_1V_1}{J}\log_e\frac{V_2}{V_1}$	$RT_1\log_e\frac{V_2}{V_1}$ or $P_1V_1\log_e\frac{V_2}{V_1}$
Adiabatic constant entropy: $n = k$ **No heat exchanges**	$PV^k = C$	$P_2 = P_1\left(\frac{V_1}{V_2}\right)^k$ $P_2 = P_1\left(\frac{T_2}{T_1}\right)^{\frac{k}{k-1}}$ Decreases	$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{k}}$ $V_2 = V_1\left(\frac{T_1}{T_2}\right)^{\frac{1}{k-1}}$ Increases	$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1}$ $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$ Decreases	None	$\frac{P_1V_1 - P_2V_2}{k - 1}$ or $c_vJ(T_1 - T_2)$
Isobaric: $n = 0$ $P = C$	$P = C$ $V = CT$	$P_2 = P_1$ No change	$V_2 = V_1\frac{T_2}{T_1}$ Increases	$T_2 = T_1\frac{V_2}{V_1}$ Increases	$c_p(T_2 - T_1)$ or $\frac{P_2k(V_2 - V_1)}{(k - 1)J}$	$P_1(V_2 - V_1)$ or $R(T_2 - T_1)$
Charles's Law: $n = \infty$ $V = C$	$V = C$ $P = CT$	$P_2 = \frac{P_1T_2}{T_1}$ Increases	$V_2 = V_1$ No change	$T_2 = \frac{P_2T_1}{P_1}$ Increases	$c_v(T_2 - T_1)$ or $\frac{V_1(P_2 - P_1)}{(k - 1)J}$	None
Polytropic: $n = \frac{r - 1 + a}{a}$ **External work = a loss in intrinsic energy**	$PV^n = C$	$P_2 = P_1\left(\frac{V_1}{V_2}\right)^n$ $P_2 = P_1\left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}}$ Decreases	$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$ $V_2 = V_1\left(\frac{T_1}{T_2}\right)^{\frac{n-1}{n}}$ Increases	$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{n-1}$ $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$ Decreases	$(P_1V_1 - P_2V_2)\left(\frac{1}{n - 1} - \frac{1}{k - 1}\right)$ or $c_v(T_1 - T_2)\left(\frac{k - 1}{n - 1} - 1\right)$	$\frac{P_1V_1 - P_2V_2}{n - 1}$ or $\frac{R(T_1 - T_2)}{n - 1}$

Values of constants: $R = \frac{1,544}{M}$, (M = molecular weight). $k = 1.4$ for diatomic gases; $k = 1.3$ for triatomic gases (see Table XIX, page 233).

The common methods for measuring quality are demonstrated in Experiment 54, pages 238 to 245. Steam tables and diagrams are given in Appendix, pages 473 to 479.

Properties of Atmospheric Air.—Atmospheric air contains, in addition to oxygen and the inert gases, smaller amounts of water vapor, solid particles, and active deleterious gases such as sulphur dioxide.

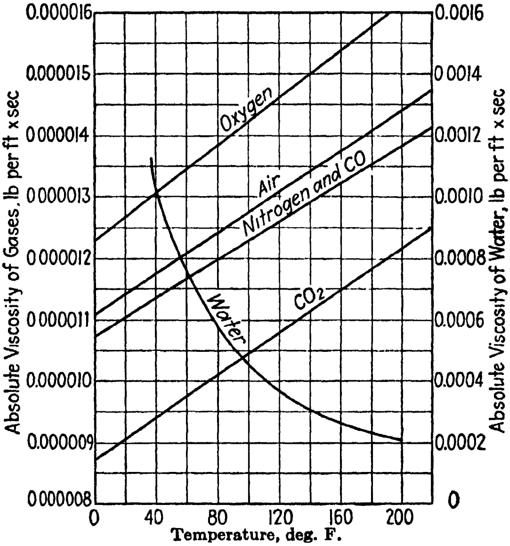

FIG. 138.—Absolute viscosities of water, air, and common gases.

NOTE: This diagram shows viscosities in lb/ft × sec or pdl × sec/sq ft as given in the A.S.M.E. Code. For slugs/ft × sec divide by 32.2.

The engineer is concerned with the measurement and control of these lesser constituents, and Experiment 55, pages 245 to 252, deals with the measurement of atmospheric moisture or humidity.

EXPERIMENT 53. Pressure-temperature Relation for Saturated Steam

I. OBJECT

It is the purpose of this experiment to investigate the changes in the temperature of saturated steam accompanying changes in pressure.

II. PREPARATIONS

Apparatus.—An apparatus is provided which consists of a small steam drum fitted with a high-pressure steam supply and with a drain or exhaust connection. If pressures below atmosphere are to be determined, the exhaust from the drum should lead to a steam condenser fitted with a vacuum pump for preserving the vacuum. Thermometer wells are inserted in the drum, and connections are provided for pressure and vacuum gages and balance dead-weight gage tester (see Fig. 54, page 85, and Experiment 2, page 27, for method of operation of this apparatus). Both steam supply and exhaust are controlled by needle valves in order that fine regulation may be obtained. The drum may be filled with steam at any desired pressure or vacuum and this pressure held constant while readings are taken. The quality of the incoming steam is varied by means of a water jacket on the steam-supply pipe.

The instruments required are a 100-lb pressure gage, a vacuum gage or mercury column, and two 400° thermometers.

III. INSTRUCTIONS

1. Test Procedure.—Check the two thermometers against each other in violently boiling water, or in a steam bath (see Fig. 32, page 51). If they do not agree, secure other thermometers. Record this temperature as the boiling point or saturation temperature for the pressure shown by the barometer. (NOTE: Most thermometers are graduated for total immersion, and an emergent stem thermometer should be attached and correction applied as determined from Fig. 18, page 36.)

Use the pressure gage merely as an indicator if dead-weight tester is used; otherwise calibrate the gage and use it as the standard. Insert both thermometers, having previously filled the thermometer wells with mercury or oil. Start condenser pump and turn on cooling water. Open both steam and exhaust valves and allow apparatus to warm up. Close steam valve until it is just "cracked," then throttle exhaust valve until the gage reads 5 lb. Hold this pressure steady by manipulating exhaust needle valve, and at the end of 2 min read both thermometers. At the end of the third minute read thermometers again, and, if these readings check with the earlier ones, record them as final. Obtain corresponding readings at 5-lb pressure increments to full line pressure.

Replace steam gage with a mercury manometer or a *calibrated* vacuum gage and take corresponding readings of vacuum and temperature to the maximum vacuum obtainable, using increments of 5 in. of mercury.

At two or more pressures during these runs, try the effect on the temperature of increasing the moisture content of the supply steam by admitting more water to the water jacket.

2. Report.—Carefully plot the pressure-temperature curve, using average temperature read for each pressure (including atmospheric boiling point). Plot corrected temperature on ordinates and absolute pressures on abscissas. Plot a similar curve from the steam tables, using a different symbol for these points. Discuss the agreement of these curves, also the accuracy of the experimental determinations and the methods of improving this accuracy. What effect does the quality of steam have on its temperature?

IV. PRECAUTIONS

1. Be sure that the steam supply is not superheated by throttling. To guard against this the water jacket had best be used during all low-pressure and vacuum runs.
2. Do not allow water to collect in the bottom of the steam drum. A steam trap may be attached to the drain line in order to guard against this possibility.
3. Correct thermometer readings for stem emergence by Fig. 18, page 36.
4. To secure certain pressures it may be necessary to open the atmospheric valve on the condenser.

EXPERIMENT 54. Steam Calorimetry

I. PREFACE

In steam generation the water surface is so violently agitated, owing to the rapid rising of steam bubbles through the disengagement area, that suspended water is entrained and wet steam is formed. The percentage of entrained moisture will increase with the rate of steam formation. On leaving the boiler a further increase in moisture will result, owing to heat losses by conduction and radiation.

It is necessary in the testing of steam equipment to determine the quality of the steam supplied. The water in the steam acts as inert material (*i.e.*, it does not contribute any work; rather it is detrimental in most cases) in passing through a steam-using unit and should not be charged against the unit in an economy test.

The devices used in determining the percentage of suspended moisture in steam are called **steam calorimeters.**

Steam quality may be defined as the percentage by weight of dry saturated steam in a given sample. It may also be expressed in the following forms:

$$\text{Steam quality} = x = \frac{\text{heat of wet steam formation}}{\text{heat of dry steam formation}}$$

$$= \frac{\text{total heat in 1 lb sample steam} - \text{heat of liquid}}{\text{latent heat of evaporation of 1 lb dry saturated steam}}. \qquad (40)$$

All values are determined at the absolute pressure of the sample.

Classification of Calorimeters.—Calorimeters may be classified as (1) superheating or evaporating, of which the throttling and electric types are examples; (2) mechanical, of which the separating is an example; (3) condensing, such as the barrel calorimeter; (4) chemical. The throttling and separating types only are treated here, since the others are seldom used.[1]

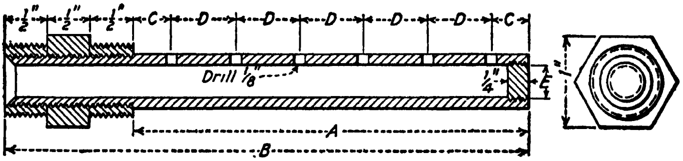

Fig. 139.—Steam calorimeter sampling tube.

Steam-sampling Nozzle.—Since only a small sample of the total steam is passed through the steam calorimeter, it is essential that the sample be representative.

The Code specifications for sampling nozzles are as follows: For pressures above atmospheric, use ¼-in. or ⅜-in. pipe, preferably the smaller. The part of the nozzle to be placed inside the main should extend within ¼ in. of the far wall and should be drilled in a straight line with ⅛-in. holes spaced ½ or ¾ in. apart. The outside end of the sampling tube should be threaded to fit the bushing to which the calorimeter or intermediate piping is to be attached (see Fig. 139). The nozzle should be installed in the line in such a position that the holes will directly face the steam flow. The ideal location is a pipe in which the steam flow is vertically downward. The second choice is a pipe in which the steam ascends vertically; at low velocities, results are likely to be erratic. Avoid securing sample from horizontal pipe if possible.

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[1] For a description of several calorimeters see A.S.M.E. Code, Instruments and Apparatus, Part II, "Determination of Quality of Steam."

The throttling calorimeter depends for its action upon the pressure drop secured by means of restricted flow through an orifice (Figs. 140 and 141).

The instrument of Fig. 140 is composed of two chambers. The inner one is supplied by steam from orifice *A*. The outer one, which serves as a steam jacket, is supplied with steam issuing through the drilled hole *B*. High-pressure steam passing through the orifice *A* expands down to the pressure in the inner chamber, which is open to the atmosphere.

If the calorimeter does not blow steam freely, the orifice is probably stopped with scale or sediment.

In the calorimeter of Fig. 141, the upstream thermometer is sometimes replaced by a steam gage. Where two thermometers are used, it is a good practice to interchange them during the test as a check on their accuracy. Fill the thermometer cup with steam-cylinder oil so as to expose the least possible length of thermometer stem. Care should be exercised to see that the cup is entirely free from water before introducing the oil; otherwise the oil and also the thermometer (when it fits closely) may be forcibly ejected when the confined water at the base of the cup changes to steam and creates pressure.

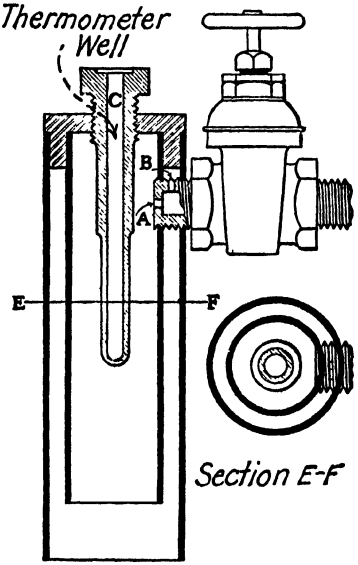

Fig. 140.—Steam-jacketed throttling calorimeter.

For the type represented in Fig. 142, the absolute back pressure is atmospheric, plus the resistance pressure created by the restricted or tortuous exit from the low-pressure side of the orifice. This excess pressure is measured in inches of mercury by a manometer.

The Ellison U-path throttling calorimeter (Fig. 143) is a commercial type having a number of advantages in design and operation. These advantages will be recognized from the following description and instructions for use. Steam path: After entering the sampling pipe, the steam in this instrument flows horizontally, with no upturns to the throttling plug, in the orifice of which it is reduced to atmospheric pressure. Immediately below the orifice, the steam impinges against a plane surface where it is divided into four atomized streams, meeting again and mixing below the plug. The throttled steam in the low-pressure steam chamber flows in a U-path, down one side and up the other, escaping at the top, where it turns and flows downward through the steam jacket and out at the bottom to the atmosphere.

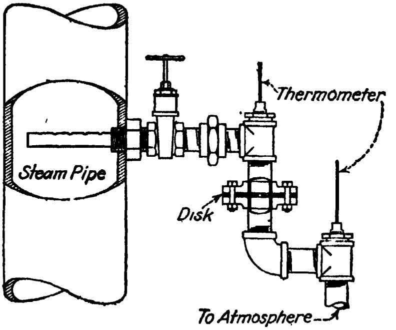

Fig. 141.—Throttling calorimeter made from pipe fittings.

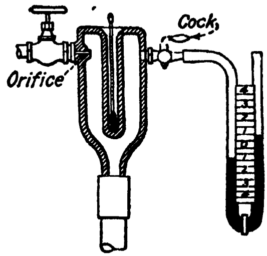

Fig. 142.—Carpenter throttling calorimeter.

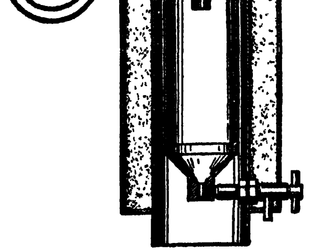

Fig. 143.—Ellison U-path throttling calorimeter.

Analysis.—Assuming that a throttling calorimeter is well lagged or steam-jacketed so that radiation is practically eliminated; then, since no external work is performed during expansion, the heat liberated by the expansion (compare total heat in 1 lb of high- and low-pressure dry saturated steam) is utilized in drying out any moisture that might have been present in the original steam sample. If more heat is liberated than is required for the vaporization of the moisture, the excess heat raises the entire body of steam to a temperature above that which corresponds to the saturation temperature at any given pressure, or it is superheated. The superheat is read by the thermometer. The instrument is limited in the percentage of moisture it is capable of handling, the maximum percentage depending upon the pressure range of expansion.

Since the final steam velocity is practically zero in the outer chamber, we conclude that the enthalpies of steam on the inlet and outlet sides of the orifice are equal. For initially wet steam, this equality may be expressed algebraically as follows:

$$xL_1 + h_1 = H_2 + c_p(T_c - T_2), \tag{41}$$

where x = quality, or percentage of dry steam in the original sample.
P_1 = absolute pressure, pounds per square inch high pressure, inlet side.
P_2 = absolute pressure, pounds per square inch low pressure, outlet side
= manometer pressure + atmospheric or barometric pressure (converted to pounds per square inch).
L_1 = latent heat of vaporization at high pressure P_1.
h_1 = heat of liquid at high pressure P_1.
H_2 = total heat of dry saturated steam at low pressure P_2.
T_2 = Fahrenheit temperature corresponding to dry saturated steam at low pressure P_2.
T_c = actual Fahrenheit temperature of low-pressure steam in calorimeter.
c_p = specific heat of superheated steam at about the constant pressure of 1 atmosphere. (Approximate value: 0.47. See steam table, page 477.)

By the use of saturated-steam tables in connection with the high- and low-pressure and temperature readings, all the values in the above equation may be obtained, excepting the value of x, which may then be calculated. The total heat on the outlet side may be taken directly from the superheated-steam tables if desired. **Throttling calorimeter problems may readily be solved by following a horizontal line on the Mollier diagram** (Fig. 195).

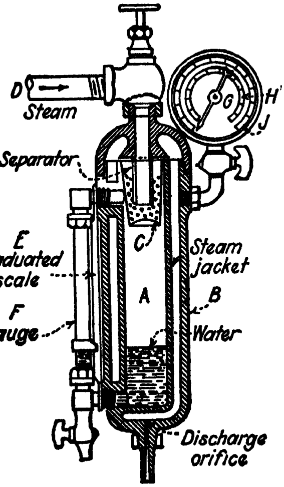

FIG. 144.—Carpenter separating calorimeter.

The Separating Calorimeter.—This instrument effects a mechanical separation of the water from the wet-steam sample by utilizing the laws of inertia and velocity reduction.

The instrument (Fig. 144) consists of two parts, an inner chamber A and an outer chamber B. The inner chamber is provided with a perforated cup C at its upper end, into which the main live-steam sample is introduced. The initial wet steam entering through pipe D is blown downward against the convex base of the cup, which causes the water content, owing to its greater weight, to be blown outward and under the lower set of inward projecting fins (which make an angle of almost 180 deg with the direction of the incoming steam) into the space surrounding the cup. Once outside the cup, the water, owing to its own weight, drops into the reservoir below.

The quantity of water trapped out may be read directly in hundredths of a pound on the calibrated scale *E* affixed to the glass water column *F* of the reservoir. The pressure gage has two graduated scales as noted. The inner scale records the pressure in the instrument, the outer scale indicates the weight in pounds of the dry saturated steam flowing through the calibrated orifice at the base of the outer chamber in a 10-min period.

The quality formula may be written:

$$X = \frac{W \times 100}{w + W}. \tag{42}$$

W equals outer scale reading, pounds of steam flowing through orifice in 10 min; *w* equals reading of weight scale, pounds of water trapped in 10 min; *X* equals percentage by weight of steam in sample.

II. PREPARATIONS

1. Apparatus.—Secure 400° short-stem throttling calorimeter thermometers, calibrated bourdon gage of suitable range, and a small mercury manometer.

2. Setup.—Install calorimeters on steam lines, using standard sampling nozzles. Blow out connections before attaching calorimeters, as scale or sediment is likely to block the small orifices. Attach discharge of separating calorimeter to a small surface condenser, to permit checking of steam-flow gage.

III. INSTRUCTIONS

1. Procedure.—Turn steam into each calorimeter slowly, then open valves wide. Allow instruments to warm up by operating about 10 min before any readings are taken. **Precaution:** Rough treatment of the gage glass on the separating calorimeter or sudden temperature changes may cause it to explode.

2. Determinations.—Make two 20-min runs under normal conditions, operating all calorimeters simultaneously. Take readings every 5 min.

Determine the effect on the readings of the throttling calorimeter of: (*a*) Reducing the inlet steam pressure to one-half its original value, by throttling the inlet valve. (*b*) Increasing the back pressure by throttling the discharge valve. Explain results of these pressure changes.

3. Results.—Fill in Form 54, and give sample calculations. Determine the quality by the use of the Mollier diagram. Discuss comparative results with the different types of calorimeters.

Form 54.—Steam Calorimeters

Barometer = ...in. mercury = ...lb per sq in.

Throttling Steam Calorimeters

Type: ....

Number	P_1	P_2	T_1	T_2	T_s	h_1	L_1	H_2	c_p	$T_c - T_2$	x
1											
2											
3											
4											

Separating Steam Calorimeters

Log of observed and computed data

Type: Carpenter standard

Number	Pressure		From instrument		From weighing		X by	
	Gage	Absolute	w	W	w	W	Instrument	Weighing
1								
2								
3								

Date.................... Observers:.... ....

....

....

IV. NOTES

In practice it has been found that when the superheat amounts to only 5 or 10°, the accuracy of a throttling calorimeter is in doubt.

The connection from the steam main to the calorimeter should be made as short as possible, at a point where there is no chance of local water pockets or temporary sprays, and at some distance from abrupt bends or obstructions to flow. When an engine is being tested, the calorimeter should be as near the throttle valve as possible, and, when a boiler is under test, the calorimeter should be as near to the boiler proper as is practicable. Steam quality varies greatly in different portions of the same pipe, much more in a horizontal than in a vertical pipe and also more when steam is descending than when ascending in a vertical pipe. Best practice therefore sanctions obtaining of steam samples from a vertical rather than a horizontal pipe.

The average quality from boilers that are run at normal capacity and with water level maintained at proper height is about 98 per cent.

A steam jet issuing into the air and producing a transparent or grayish-white color close to the orifice and an even cutting sound may be said to be dry. If white predominates and the sound is cutting with no appreciable sputtering, then the steam may be judged to contain less than 2 per cent of moisture. When sputtering is plainly discernible, more than 2 per cent of moisture is present. Superheated steam feels dry to the skin when the hand is passed quickly through a jet. Wet steam will sting and burn the skin.

For a well-insulated throttling calorimeter, no allowance will normally need to be made for radiation if 200 lb or more of steam pass the orifice per hour. The insulation should be 1 to 2 in. thick and should be applied to all exposed parts of the calorimeter and sampling connection.

Combined calorimeters consisting of the separating and throttling types connected in series, with the steam entering the separating calorimeter first, will permit of moisture determinations even when the steam pressure is very close to atmospheric pressure (see A.S.M.E. Code, page 11).

EXPERIMENT 55. Humidity and Its Determination

I. PREFACE

The humidity, or invisible water-vapor content of atmospheric air, is from an engineering point of view a very important constituent. The properties of wood, paper, cotton, silk, and other of the fibrous materials of industry are sensitive to the quantity of moisture in the atmosphere. Hence in industries dealing with these and many other materials such as chemical salts, tobacco, pastes and doughs, paints and varnishes, the engineer is called upon to measure and to regulate the atmospheric humidity in the storage spaces and workrooms.

Humidity, temperature, air motion, and radiation determine human comfort in a given atmosphere.

These facts have given rise to a typically American development, **air conditioning,** or the engineering control of indoor air conditions. Much is expected of this new field of engineering, because it is assumed that favorable living and working conditions are as fundamental to human well-being as is for instance good transportation.

It is evident that as a basis of progress in air conditioning the engineer must be able to *measure* the humidity, temperature, dust content, and motion of the air, and also the presence of lesser impurities and odors.

II. THEORY

Humid air is essentially a mixture of dry air and steam, and the following terms are used in describing this mixture.

Saturated air is a mixture of air and saturated steam and contains all the moisture it is possible for it to hold without showing fog (compare definition of saturated steam). The weight of the water vapor in saturated air depends on its temperature only and may be read from the saturated-steam tables (density or specific volume).

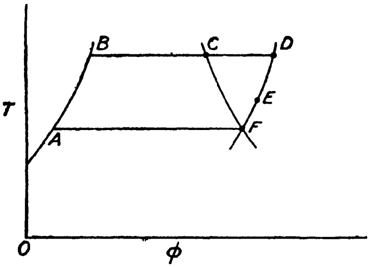

FIG. 145.—Temperature-entropy diagram for moisture in air.

Relative humidity may be defined in two ways: (1) It is the ratio of the actual partial pressure of the vapor to the saturation pressure at the dry-bulb temperature. (2) It is the ratio of the actual density of the vapor to the density of saturated vapor at the dry-bulb temperature.

In Fig. 145, *D* represents the actual condition of the vapor in the air mixture, and the two definitions of relative humidity may be equated as follows:

$$\frac{P_D}{P_C} = \frac{\rho_D}{\rho_C}, \tag{43}$$

where P represents pressure, and ρ represents vapor density in pounds per cubic foot. Since density is the reciprocal of specific volume, Eq. (43) will be recognized as a statement of Boyle's law, applying to the isothermal *CD* and treating the superheated steam as an ideal gas.

Specific humidity is the weight of water vapor associated with 1 lb of dry air, usually expressed as grains of moisture per pound of dry air. This quantity is sometimes called the *absolute* *humidity* (A.S.M.E. Code). Specific humidity depends on the dew point only, and can be obtained from Table XLV, Appendix, page 480, or from a psychrometric chart (Fig. 146).

Vapor density, the weight of water vapor per cubic foot (pounds or grains), is another useful quantity. Vapor densities of saturated air may be obtained from the saturated-steam tables (pages 473 and 480).

Dew point is the temperature at which condensation begins, when the air mixture is cooled at constant pressure (point F, Fig. 145).

Wet-bulb temperature is the temperature that water will assume owing to evaporation in air, if its surface area is large compared with its mass (point E, Fig. 145). A moistened wick or fabric sheath covering a thermometer bulb is used to obtain the wet-bulb reading, but *the wet-bulb temperature is also attained by a water spray.* An insulated, saturating air washer or humidifier, with recirculating spray water, produces the **temperature of adiabatic saturation** which is for engineering purposes the same as the wet-bulb temperature.

Enthalpy of the air mixture is defined as the enthalpy of the dry air above 0°F plus the enthalpy of the vapor above 32°F:

$$h = 0.24t + Wh_s, \tag{44}$$

where t is the dry-bulb temperature, W is the specific humidity in pounds of vapor per pound of dry air, and h_s is the enthalpy of superheated steam at the dry-bulb temperature. The Carrier **total heat** content (Σ) differs from the true enthalpy in that it does not include the enthalpy of the liquid, but there is very little difference in the final numerical values. Since air remains at a constant wet-bulb temperature when it is saturated adiabatically, either *total heat or enthalpy may be obtained from a table or chart at the wet-bulb temperature.* The psychrometric chart Fig. 146 gives total heat content, Σ, but Table XLV, page 480, gives enthalpy.

The most commonly used psychrometric formula is the one proposed by W. H. Carrier:

$$\frac{\text{Relative humidity}}{100} = \frac{P_w}{P_d} - \frac{(P_b - P_w)(t_d - t_w)}{(2{,}800 - 1.3t_w)P_d}, \tag{45}$$

in which P_d and P_w are the absolute pressures in inches of mercury

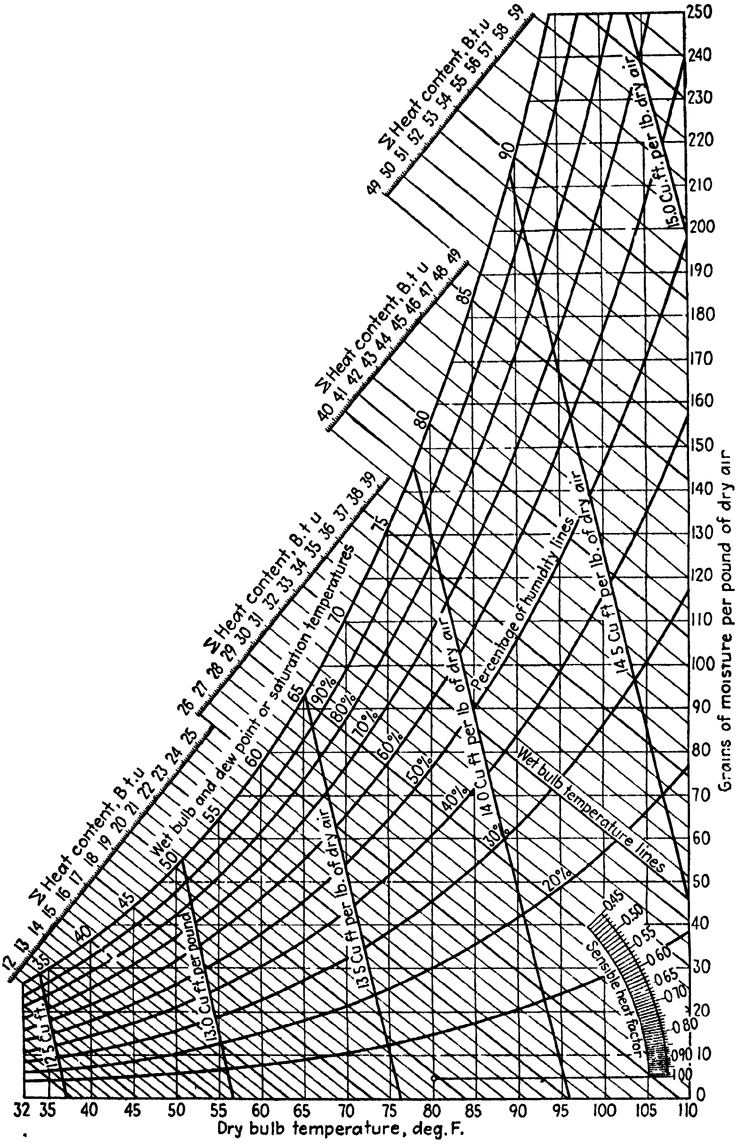

FIG. 146.—Psychrometric chart for 29.92-in. barometer. (*Carrier and Mackey.*)

for saturated steam at the temperatures t_d and t_w; P_b is the barometric reading in inches of mercury; and t_d and t_w are the dry-bulb and wet-bulb temperatures, respectively. A psychrometric chart based on this equation and on the steam tables, is shown in Fig. 146.

III. INSTRUCTIONS

1. Determinations.—The purpose of this experiment is to compare various humidity instruments in actual use, and it will be assumed that one of each of the following types of instruments is available: (1) sling psychrometer; (2) aspiration psychrometer; (3) stationary-bulb psychrometer; (4) hair or fiber hygrometer.

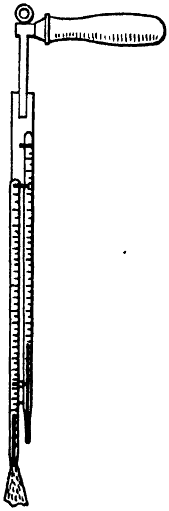

FIG. 147.—Sling psychrometer.

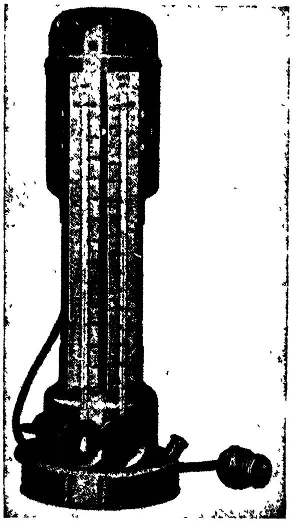

FIG. 148.—Aspiration psychrometer.

Take at least three readings of each instrument in two or more rooms and outdoors as designated by the instructor. Reject and repeat any readings that do not check.

Use the stationary-bulb psychrometer in air currents of different velocities, obtaining at least four readings in the range below 2,000 ft per min. The air stream may be obtained from some type of fan or blower and velocity measured by an anemometer.

Report the following for all readings: dry-bulb temperature, wet-bulb temperature, relative humidity, absolute humidity in grains per cubic foot, absolute humidity in grains of moisture per pound of dry air, dew point, partial pressure of water vapor, partial pressure of air, heat content above 0°F. Use psychrometric tables, Table XLV, page 480; check by large chart if available; otherwise use Fig. 146, page 248, or Eq. (45) and steam tables, page 480. Determine and report on the location of each test point relative to the "comfort zone" (Fig. 196, page 481). Discuss relative accuracy of the various determinations.

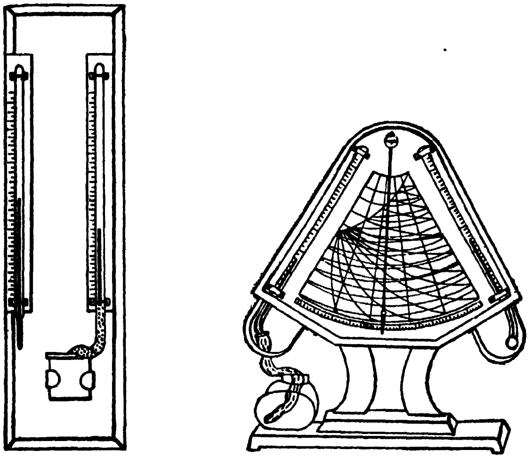

FIG. 149.—(*a*) Hygrometer, (*b*) Hygrodeik.

2. Instruments and Their Use.—The *sling psychrometer* (Fig. 147) is the "official" instrument recognized by the U. S. Weather Bureau and the A.S.H.V.E. and A.S.M.E. test codes. To use this instrument, moisten the wet-bulb wicking only, with water at or near the wet-bulb temperature. Whirl rapidly for ½ min, then read, observing the wet-bulb reading first. Whirl, read, whirl, read, until the lowest possible wet-bulb reading is obtained. Complete duplicate determinations are necessary to ensure accuracy. The sling psychrometer is not protected from radiation, and it should not be used near a radiating body.

The *aspiration psychrometer* (Fig. 148) is protected from radiation and affords the most accurate practical method for measurement of humidity. This instrument may or may not have a built-in reservoir.

The *stationary psychrometer* or hygrometer (Fig. 149) usually has a water reservoir. Readings of these instruments are affected by air currents, and the wet-bulb depression obtained may be in error by as much as 20 per cent in some cases.[1] The wet-bulb reading is also affected by radiation. The accuracy of this instrument is much more satisfactory if it is placed in a positive air current, as in a duct or in front of a desk fan.

The *hair or fiber hygrometer* is made in a great variety of forms, utilizing human hair, silk, wood, and similar materials. Such instruments are usually less accurate than the wet-bulb types and must be calibrated by reference to the latter. A good instrument may be in error from 4 to 10 per cent or more. They have several other disadvantages such as sluggishness, variation with temperature, and change in calibration due to a "permanent set" of the material or to dust and other impurities deposited from the atmosphere. These instruments are more satisfactory for control purposes than for use in engineering tests.

Several other types of humidity instruments are available. One recent commercial instrument measures the difference in thermal conductivity between dry and humid air, using a comparison cell and a Wheatstone-bridge circuit.

The *kata thermometer* (see page 268) is widely used in England and to some extent in this country. There are three types of kata thermometers, but the same thermometer may be used in all three cases if desired. As a dry-bulb instrument, this thermometer measures cooling by radiation and convection only. If covered with a wick, the rate of cooling depends on radiation, convection, and evaporation. If the bulb is inserted in a special transparent vacuum bottle during cooling, the rate of cooling will depend almost entirely on radiation. From these three readings, the room comfort conditions, *i.e.*, temperature, humidity, air motion and radiation, may be accurately determined.

IV. NOTES

The accuracy of the wet- and dry-bulb thermometers should be noted before the wick is moistened by reading both thermometers in an air stream, when dry. In a good instrument no difference in reading should be visible.

The sling psychrometer should be rotated at a speed of not less than 15 ft per sec for consistent results (about 150 rpm).

[1] See Carrier and Lindsay, "The Temperatures of Evaporation of Water into Air," *Trans. A.S.M.E.*, 1924; see also A.S.M.E. Code, p. 16.

Use clean water; distilled water is preferred. Do not touch the wet bulb, as the slightest oily deposit on the wicking will cause errors.

The water used for moistening the wick should not be more than 10° from the wet-bulb temperature.

If the barometer reading is not close to standard, special charts should be used (see discussion in A.S.M.E. Code).

The wet-bulb error is always positive and is larger at low humidities.

The dry-bulb temperature is a measure of sensible heat, the wet-bulb temperature a measure of total heat, and the dew-point temperature a measure of moisture content or absolute humidity.

There are several types of recording humidity instruments which make a continuous record either of wet- and dry-bulb temperatures or of relative humidity directly. Bourdon-tube thermometers, resistance thermometers, or thermocouples may be used. The Weather Bureau standard recording instrument is a hair hygrometer. These types of units are also used in humidity-control devices.

Humidity measurements for temperatures below freezing may be made in the usual way. Additional time (about 5 min.) is usually necessary, and special care must be taken to secure lowest possible wet-bulb reading. (Tables for low temperature work are given in References 3, 4 and 5, following.)

V. REFERENCES

1. "Humidity Determinations," A.S.M.E. Test Code on Instruments and Apparatus, Part 18.
2. Behar, M. F.: "Industrial Humidity Instruments," *Instruments*, September–November, 1930.
3. A.S.H.V.E., "Guide," (published annually).
4. U. S. Weather Bureau, *Bull.* 235.

EXPERIMENT 56. Performance of an Air-conditioning Unit

I. PREFACE

The rapid growth and public acceptance of air conditioning have resulted in a rather loose usage of the term. In general, the functions of heating, cooling, cleaning or washing, humidifying and dehumidifying, and air distribution should all be included in the complete process of air conditioning. In this experiment it will be assumed that a spray type unit is available, with surface coils for additional heating or cooling and with either heat or refrigeration available for application to the spray water.

A typical air-conditioning unit is shown diagrammatically in Fig. 150. When outdoor air is being conditioned in winter, it is preheated in a "tempering coil," which may be by-passed when it is not required. Provision may be made for recirculated air to be mixed with the outdoor supply. A separate heating or cooling unit for the spray water may be used, or this coil may be located in the reservoir below the spray nozzles, or the steam may be introduced directly into the water. The "eliminator" plates are staggered, zigzag plates designed to eliminate water drops from the air leaving the washer. Extra water-flooded "scrubber" plates are sometimes used to assist in the air cleaning.

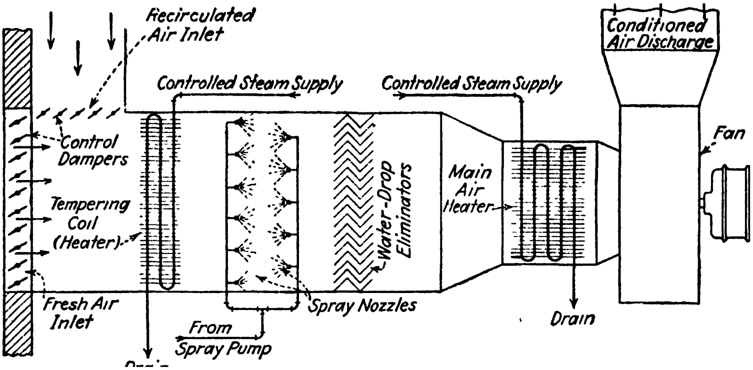

FIG. 150.—Diagram of air-conditioning unit.

The main heater is usually an extended-surface unit, under thermostat control. The temperature of the exit air may also be controlled by providing a by-pass damper under thermostat control, so that only enough air is put through the heater to maintain the desired exit temperature after mixing.

II. INSTRUCTIONS

1. Determinations.—The tests it is possible to run will depend both on the equipment available and upon the prevailing weather conditions. Wherever possible, **three separate tests** should be made.

(1) *Adiabatic cooling* test, or operation of the spray only, with no heat or refrigeration. This process is valuable for increasing the comfort conditions in hot, dry climates, but its limitations should be clearly understood. The spray water tends to assume the wet-bulb temperature of the entering air, and, if a large spray capacity is available, the air will leave in a practically saturated

condition at the original wet-bulb temperature. (2) *Heating and humidifying* are the usual winter conditioning, and this part of the experiment should be performed with outdoor air supply and with heat supplied to the spray water. Tempering and reheating coils may be operated or not, as directed by the instructor. (3) *Cooling and dehumidifying* are the required summer air conditioning in most of the United States, and for successful operation of this part of the test the air must be supplied to the unit in a warm and humid condition. This condition may be produced in the laboratory in winter by allowing steam to escape into the air at the inlet of the test unit. The spray water temperature must be below the dew point of the entering air mixture in order to produce dehumidification.

2. Readings.—Take dry-bulb and wet-bulb readings at the inlet and outlet of each unit in the system. To obtain good averages it may be necessary to use mixing baffles or to take multiple readings at some locations. Observe precautions given on pages 251 and 256. Measure air quantity by orifice, pitot tube, or anemometer (Chap. VII). Measure water and steam quantities and temperatures. Water temperatures must be measured very accurately if heat balance is to check. Take all readings every 5 min, until 30 min of steady conditions are obtained for each test. To aid in locating errors, trace processes on psychrometric chart during each test.

3. Report.—Tabulate dry bulb, wet bulb, dew point, relative and absolute humidities, and enthalpy (or total heat content) of air for each measuring station (see Experiment 55 and psychrometric chart, page 248). A suggested tabulation of results is given in Form 56. **Show the complete process of each test** on a sketch of the psychrometric chart. Discuss the changes in "effective temperature" with relation to the comfort zone, as shown in Fig. 196.

III. NOTES

Ample time must be allowed for the establishing of steady thermal conditions. The best method is to take all readings every 5 min, continuing until at least ½ hr of steady readings have been obtained.

For strict accuracy in the determination of heat furnished by steam to surface heaters, the steam should be slightly superheated (see Radiator Test, page 216). If this is not practicable, steam separators of ample size should be located at the entrance to the heater.

Form 56.—Performance of an Air-conditioning Unit

Description of Unit:
Manufactured by..
Type and rated size..
Type of controls..
Fan: Type.........Wheel diam........Drive.......Speed...........
Spray nozzles: Type.... ....Number. .... ..Spacing...... ..
Water heater and cooler: Type and size. ....
Air heaters: Type and size ....

Air Conditions									
Location*	Adiabatic cooling			Humidifying			Dehumidifying		
	A	*B*	*C*	*A*	*B*	*C*	*A*	*B*	*C*
Wet bulb, deg. fahr..........									
Dry bulb, deg. fahr..........									
Dew point, deg. fahr.......									
Relative humidity, per cent ..									
Specific humidity, grains per lb..									
Enthalpy, B.t.u. per lb.........									
Density, lb. per cu. ft.. .									

* *A*: air entering washer. *B*: air leaving washer. *C*: air leaving heater.

Heat-balance Summary			
		Humidifying	Dehumidifying
1	Average air velocity in duct, ft. per min..		
2	Volume of air discharged, c.f.m...........		
3	Specific volume of dry air, cu. ft. per lb		
4	Weight of air discharged, lb. per min.....		
5	Heat exchanged by air in washer, B.t.u. per min............................		
6	Heat exchanged by air in heater, B.t.u. per min............................		
7	Weight of water circulated, lb. per min..		
8	Temperature change of water, deg. fahr...		
9	Heat added or removed by water, B.t.u. per min............................		
10	Steam condensed in heater, lb. per min. ..		
11	Heat added by steam, B.t.u. per min.....		

Thermometers should be checked for accuracy, or they may be interchanged periodically for this purpose. Thermocouples may be used to advantage in certain locations.

Air leaving a heater or cooler is often stratified, and the reading of a single stationary thermometer is not to be relied upon as a true average for the stream.

Heat transfer to or from the casing or duct work should be checked by using the heat-transfer coefficients of Table XVII, page 204, and this item should be included in the heat balance if it is greater than 1 or 2 per cent. In certain cases a correction may also be necessary for the heat added by the fan.

A constant water level should be maintained in the spray tank and in the heating or cooling tank if this is separate. Drains should be checked for leakage.

The operation of individual spray nozzles should be checked, as these units are likely to become clogged in systems that are intermittently operated.

CHAPTER VII

FLUID FLOW AND FLUID DYNAMICS

I. PREFACE

Applications of fluid flow are encountered by the mechanical engineer in aeronautics, hydraulics, heating and air conditioning, power plants, internal-combustion engines and several other fields. Fortunately, many of the problems of fluid dynamics may be analyzed by applying a few simple laws. The law of conservation of energy is the most useful, and problems are further simplified by making certain assumptions, such as those that neglect viscosity or neglect compressibility.

In the flow of an "ideal fluid," which is nonviscous and incompressible, the most important **forces acting upon the fluid** are those due to difference in pressure at various points and those due to the inertia or the kinetic energy of the fluid itself. Unless the flow is horizontal, gravity forces will also be involved. Empirical correction factors may be used to account for the shearing forces due to viscosity of the real fluid and for elastic forces due to compressibility. An example of this procedure is the application of the "hydraulic equation" [Eq. (52)] to an orifice meter used for air, as in Experiment 61. Here the viscosity correction is contained in the coefficient C, and the compressibility correction is the factor Y, from Table XXIII.

The lift of an airplane wing, or the force of propulsion from the sail of a ship or from the propeller of an airplane is calculated by similar analyses, using the law of conservation of energy (or conservation of momentum). The small effects due to the viscosity and the compression of the air are again taken into account by empirical factors (see Experiment 61 D).

Three important classes of experimental problems in fluid mechanics are: (1) Those dealing with *flow through pipes* and channels (Experiment 62). (2) Those dealing with the *forces* produced by relative motion of a body and an unconfined fluid, such as the *lift* and the *drag* of an airfoil (Experiment 63). (3)

Those dealing with fluid *measurement* (Experiments 57 to 61). Much of the experimental work in all of these cases has to do with the measurement of empirical factors, such as the friction factor for pipes (Fig. 151), the coefficient of discharge for orifices and nozzles (Figs. 156, 157, 158, and 160), and the drag coefficients for geometrical shapes (Fig. 202).

II. FLOW-STREAM CHARACTERISTICS

The primary analyses of flow problems are based on the hypothetical case of the *frictionless, adiabatic, steady flow of the ideal fluid.* In other words the flow involves no shearing forces due to viscosity, no elastic forces due to compressibility, no heat transfer, and no unsteady conditions. The equipment is usually assumed to be horizontal, and thus gravity forces are also cancelled.

Applications of the ideal-flow equations require the use of several empirical factors as already described, and, in order to determine these factors or even to select them from charts and tables, some of the characteristics of a fluid stream must be understood.

If a gas or a vapor is flowing in a pipe, and the average velocity of the stream is increased from zero to a maximum, **three or more types of flow** may actually be involved:

1. Viscous, laminar, or streamline flow, in which the fluid particles move in parallel paths, with decreasing velocity as the solid boundary (pipe wall) is approached. This type occurs with any fluid when the "Reynolds number"

$$\mathrm{Re} = \frac{\text{pipe diameter} \times \text{average fluid velocity}}{\text{kinematic viscosity of the fluid}} = \frac{DV}{\mu/\rho} \qquad (46)$$

is below a certain "critical" range.

2. Turbulent flow, in which the fluid particles move not only parallel with the stream but also in random turbulent paths across the stream. (The result is somewhat like the "rapids" of a river.) This type occurs with any fluid when the Reynolds number exceeds an upper critical value of about 2,000 to 3,000.

3. Supersonic flow, or flow in which the "acoustic velocity" or velocity of sound in the fluid is exceeded. This occurs only in the case of gases and vapors when a large pressure drop is allowed, *i.e.*, when a large amount of potential energy is quickly converted into kinetic energy, as in a turbine nozzle. The velocity of sound in a perfect gas is $V = \sqrt{gkp_t/\rho_t}$. When the first derivative of the quantity under the radical in Eq. (55), page 269, is equated to zero, it is found that the maximum flow occurs when

$$\frac{P_2}{P_1} = \left(\frac{2}{1+k}\right)^{\frac{k}{k-1}}. \tag{47}$$

The same rate of flow also occurs for *any* value of the pressure ratio less than that given by Eq. (47), *i.e.*, **the flow quantity is unaffected by the downstream pressure if the latter is below the critical value.** Hence this third type is often called "unaffected flow." Throughout this range the velocity beyond the throat exceeds the acoustic velocity. The *critical pressure ratio* **for air** is about **0.53,** for **superheated steam** about **0.545**, and for **saturated steam** about **0.58.** The first two types of flow, *i.e.*, viscous and turbulent, are also obtained with a liquid, as the velocity of flow is increased, or rather, as the Reynolds number is changed.

III. THE REYNOLDS NUMBER

The **Reynolds number,** $DV\rho/\mu$, is an index or criterion showing the *ratio of the inertia forces to the viscous forces.* As the viscous forces are increased, this ratio becomes smaller, and vice versa. When using the Reynolds number as an index or parameter [as when an orifice coefficient is plotted against it (Fig. 157)], **dynamic similarity** must be maintained. This requires that throughout the range of sizes, the entire equipment must be kept *geometrically similar.*

Reynolds number is used as the abscissa for plotting the friction factor for pipes (Fig. 151), the drag coefficient for a sphere or cylinder (Fig. 202), and many other empirical factors. Hence it becomes very important to evaluate this dimensionless quantity, $DV\rho/\mu$, for any particular case. The value D may be any significant linear dimension, but it can usually be reduced to a diameter or to an equivalent diameter. Consistent units must be used; hence for the foot-pound-second system, **D is in feet,** the average velocity in the direction of the stream **V is in feet per second,** and the density ρ **is in pounds per cubic foot.** The absolute viscosity μ **must be in English units,** and its dimensions are pounds per second-foot (see pages 163–166 for definitions and conversion factors for viscosity). Since engineering tables usually give density as weight per unit volume (pounds per cubic foot), the *gravitational units must be used throughout.* If density is given as mass per unit volume (slugs per cubic foot), this value must be multiplied by g (32.2). **The following examples illustrate the calculation of the Reynolds number.**

Example 1.—Calculate the Reynolds number for water at 70°F flowing through a standard 2-in. steel pipe at an average velocity of 5 ft per sec.

Solution.—From the steam tables (page 473) the weight-density of water at 70°F is 1/0.01606 = 62.3 lb per cu ft. The viscosity of water at 70° from Table XL, page 469 is 1.0 centipoise. Converting this to English units (page 165) it becomes 1.0 × 0.000672 = 0.000672 lb/sec × ft. (See also Fig. 138, page 236.) From Table XLII, page 472, the internal diameter of 2-in. standard pipe is 2.067 in. Then

$$\mathrm{Re} = \frac{DV\rho}{\mu} = \left(\frac{2.067}{12}\right) \times 5.0 \times \left(\frac{62.3}{0.000672}\right) = 79{,}800.$$

Example 2.—Calculate the Reynolds number for air at 70°F and 14.7 lb abs pressure, flowing in a 1-in. diameter tube with an average velocity of 44 ft per sec.

Solution.—The weight of 1 cu ft of air at 70°F and 14.7 lb abs is

$$W = \frac{PV}{RT} = \frac{14.7 \times 144 \times 1}{53.3 \times 530} = 0.0749 \text{ lb per cu ft.}$$

The viscosity of air, from Table XIX, page 233, is approximately 0.018 centipoise or 0.018 × 0.000672 = 0.0000121 lb/sec × ft. Then

$$\mathrm{Re} = \frac{DV\rho}{\mu} = \frac{1}{12} \times \frac{44 \times 0.0749}{0.0000121} = 22{,}700.$$

The same solution would be obtained by using the absolute viscosity of air as given in Fig. 138, page 236, $\mu = 0.0000122$ lb/sec × ft.

IV. FLUID FRICTION, "DRAG" AND "LIFT"

The flow of fluids through pipes, fittings, valves, etc., results in a friction loss or drop in pressure. Similarly, the motion of a body within a fluid, such as the motion of a ship in the water, is resisted by a frictional "drag." If the axis of a moving body is not parallel to the direction of motion, a cross-stream component is also produced. This force is the "lift" of an airplane wing. It is also the force that propels a sailboat.

Both pipe friction and drag are produced partly by the viscous forces or skin friction and partly by the dissipation of energy in turbulence and vortex action. When the viscous forces predominate, *i.e.*, below the critical Reynolds number, the pressure drop in a pipe is directly proportional to the velocity. When inertia forces predominate (Re is greater than 3,000 approx), the friction loss is proportional to the initial kinetic energy of the stream, *i.e.*, to the average velocity pressure $V^2/2g$, or to the *square* of the velocity instead of to the first power. As might be expected from the analogies of flow of electricity and heat

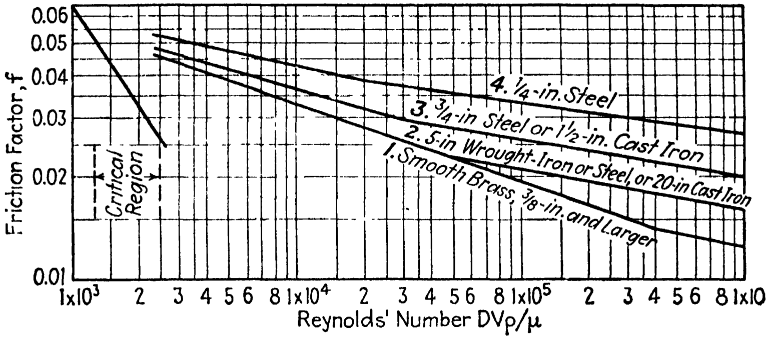

Fig. 151. —Friction factors for commercial pipe. (*From R. J. S. Pigott, "The Flow of Fluids in Closed Conduits," Mech. Eng., August,* 1933, *p.* 497.)

[Eq. (27), page 200], the pressure drop is directly proportional to the length of the conduit and inversely proportional to its area. Hence the **common equation for pipe friction**

$$\Delta h = \frac{\Delta P}{\rho} = f \frac{L}{D} \frac{V^2}{2g}, \tag{48}$$

or

$$P_f = 0.000108 \rho f \frac{L}{D} V^2, \tag{49}$$

where P_f = pressure drop due to friction, pounds per square inch.
L = length of pipe, feet.
ρ = fluid density or specific weight, *pounds* per cubic foot (any fluid, liquid, gas, or vapor).
V = mean fluid velocity, feet per second.
D = inside pipe diameter, feet.
f = friction factor, a function of Reynolds number.

The friction factor may be obtained from Fig. 151. This graph is a condensed summary of most of the experimental work that has been done on pipe friction. It shows four different degrees of pipe smoothness or "relative roughness" (ratio of height of surface projection/diameter of pipe.) In order to allow for the resistance of valves and fittings in the average pipe-friction problem, an equivalent length of straight pipe may be added to L [Eq. (49)]. Equivalent lengths are given in Fig. 152.

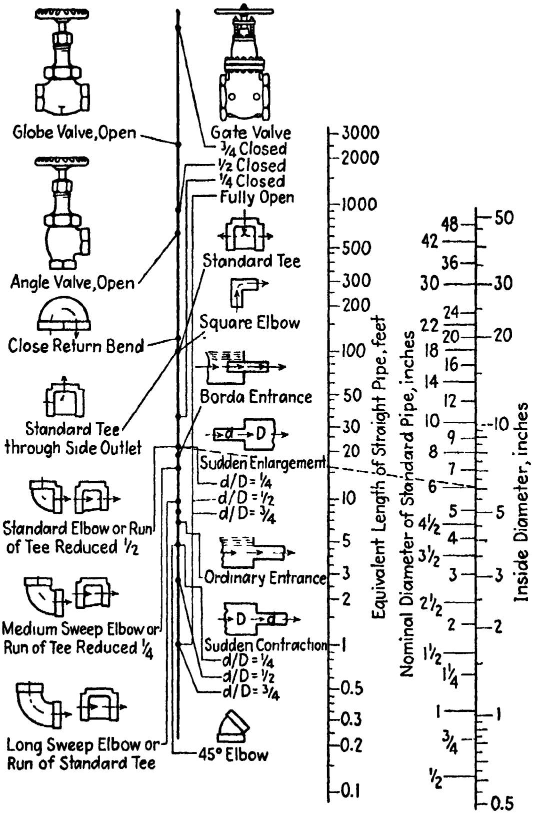

FIG. 152.—Equivalent flow resistance of valves and fittings. (*Crane Co.*)

The **drag coefficient** for a geometrical shape is complicated by many factors (see Experiment 51). It may however be represented in the same simple manner as is the pipe-friction coefficient. Figure 202, page 493, gives typical curves of the drag coefficient C_d as defined by the following equation:

$$F = C_d \rho A \frac{V^2}{2g}, \tag{50}$$

where F = total force exerted on the immersed body.
ρ = weight-density of surrounding fluid, pounds per cubic foot, (specific weight in pounds, *not* slugs).
A = projected area of the body on a plane at right angles to flow.
$V^2/2g$ = average kinetic energy (or velocity head) of approaching stream.

The **lift coefficient** for an airfoil is related to its dimensions. But the results of airfoil tests in small wind tunnels cannot be used for accurately determining the performance of modern airplanes, because of the difficulties of obtaining dynamic similarity.

V. FLUID-FLOW MEASUREMENT

1. Preface.—The measurement of fluid flow is an everyday engineering job, especially for the mechanical and the chemical engineer. Whether the fluid to be measured is steam to a turbine, cooling air for an electric generator, oil in process in a refinery, or ammonia in a refrigerating plant, the procedure is the same. Moreover, when the principles and methods applying to these cases are understood, a clear picture may easily be obtained of the working of other flow units, such for instance as an automobile carburetor, a jet pump or injector, an airplane-speed meter or a locomotive draft nozzle.

The importance of this subject is shown by the fact that ever since 1916 the A.S.M.E. has maintained a Special Research Committee on Fluid Meters, and this Committee has issued several hundred pages of publications. In their basic handbook, "Fluid Meters, Their Theory and Application, Part 1," the Committee gives a classification of fluid meters under the two main headings of (1) *quantity meters* and (2) *rate meters.* A classification of the common fluid meters is presented in Table XXI.

Fluid measurement is essentially a measurement of *quantity, i.e.*, of weight or volume. The simple methods of weight and volume measurement were treated in Chap. II, and a "quantity meter" combines one of these simple weight- or volume-measuring devices with a counting mechanism. Each successive weight or volume passing through the meter actuates the counter and thus produces an indication or record (see Experiments 6, 7, and 57). A **positive quantity meter** is the most accurate commercial fluid-measuring device, and is therefore preferred for test work.

TABLE XXI.—CLASSIFICATION OF FLUID METERS
(From Marks)

Common name or type	Fluids metered	Class of measurement	Maximum capacities	Secondary elements			
				Rate indicator	Chart recorder	Counter or totalizer	Accuracy, ± %
Division One: Quantity Meters							
Weighers and tilting traps	Liquids	Weight	Unlimited	..	..	X	1
Disk water meters	Clean liquids	Volume	20–1,000 gpm	..	..	X	2–4
Piston meters	Clean liquids	Volume	1–1,000 gpm	..	..	X	½–2
Drum condensate meters	Hot water	Volume	250–12,000 lb per hr	..	..	X	2–3
Bellows gas meters	Gases	Volume	150–17,000 cfh	..	..	X	1–3
Wet gas meters	Gases	Volume	100–100,000 cfh	..	..	X	½–2
Rotary gas meters	Gases	Volume	500–1,000,000 cfh	..	X	X	½–2
Division Two: Rate Meters (for steady flow only)							
Orifice plate	Any fluid	Head (constant area)	Unlimited	X	X	X	1–3
Flow nozzle	Any fluid	Head	Unlimited	X	X	X	1–3
Venturi meter	Any fluid	Head	Unlimited	X	X	X	1–3
Pitot tubes	Any clean fluid	Head	Unlimited	X	..	..	1–3
Weirs	Liquids (in open channels)	Head-area	Unlimited	X	X	X	½–2
Area meters (variable orifice)	Clean fluids	Area (constant head)	Unlimited	X	X	X	1–3
Velocity meters	Air or water	Velocity	Unlimited	..	..	X	2–10
Thermal meters	Air and clean gases	Weight	25,000–2,000,000 cfh	X	X	X	1–2

Most of the rate-type meters are **inferential meters,** *i.e.*, the quantity of fluid is merely "inferred" from some action of the stream on the primary element of the meter. Accurate results are more difficult to obtain with an inferential meter because the fluid may pass through the meter even when the indicating mechanism is blocked. The rate type meter has several advantages, however, which make it particularly well adapted to commercial work (see Experiment 61). Arranging the various kinds of inferential meters in the order of their importance in engineering use, the "head" meters would come first; the "head-area" or weir meters probably next; and the "current" meters third. "Area" meters and "thermal" meters are not so widely used. Most commercial flow meters are head meters, using either a thin-plate orifice, a flow nozzle, or a venturi as their primary element (Fig. 155, page 275).

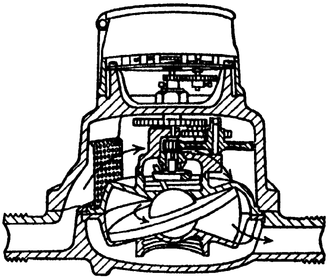
FIG. 153.—Interior of a domestic water meter, nutating disk type.

2. Rotary and Reciprocating Meters.—There are several types of enclosed rotating-element meters, in addition to the open-vane types discussed in Sec. 3, page 266 (and in Experiment 57).

Rotary-disk meters are generally used for metering cold water on domestic and commercial service lines. They consist of a circular metering chamber with a conical roof and floor (Fig. 153) divided into two equal compartments by a "nutating" disk. The disk does not rotate about its own axis, but the shaft on which it is mounted generates a cone with apex downward. Motion of the disk is guided by the two half balls mounted upon it, and with each complete revolution a fixed volume of water passes through. The circular motion of the upper end of the disk shaft operates the counting gears. The error of a disk meter will usually be 2 or 3 per cent, and they are intended for use only for clean water below 125°, though they are also furnished with special disks for hot water and other clean liquids.

Revolving-drum condensate meters are common for metering purchased steam, and they usually measure water at or near atmospheric pressure, with gravity discharge. An error of 2 or

3 per cent may be expected. Approximately 18 in. static head is required to operate the meter, and the meter must be installed so as to be self-clearing, or accuracy will be affected. Ordinary pulsations or intermittent flow will not affect the accuracy of the revolving drum meter.

Rotary displacement meters of the lobed impeller, gear, or screw type are sometimes used for water, though they are better adapted to oil and other lubricating liquids.

Piston meters are made in a variety of combinations of one, two, or more pistons, reciprocating or oscillating in fixed chambers. Piston meters are capable of a high degree of accuracy, but actual errors will depend on the type of service for which they are designed, on maintenance, and on the accuracy of adjustment.

The **bellows gas meter** is used for domestic and commercial gas metering. The gas is metered by alternate filling and emptying of two leather bellows chambers, connected to a counting mechanism (see Experiment 7). *Wet gas meters* are of the rotary-drum type and are more accurate than the dry or bellows type. Pressure and temperature corrections must be applied to the readings of all gas meters.

3. Anemometers and Aircraft-speed Instruments.—The term "anemometer" may be applied to all types of air-velocity instruments (see Experiment 57, page 285), but it usually refers to a vane-wheel or a cup type of rotating air-velocity meter. For air logs of lighter-than-air ships, both the rotating-vane and the rotating-cup instruments have been used, with electrical indicating instruments (see Fig. 165, page 285).

For **airplane-speed instruments** a pitot tube is commonly used as the primary element. This is connected to a sensitive diaphragm type differential airplane instrument, but the essentials of the impact and static openings are retained. To prevent the formation of ice on airplane pitot tubes they are specially shaped (and may be electrically heated). If the tube is swiveled or suspended instead of rigidly fixed, it is equipped with directive vanes. Vertical tubing and drip traps prevent stoppage by rain. Venturi heads have been used on some airplane-speed meters, but these are not favored by American engineers. All speed meters are subject to errors of position (yaw and inclination), to interference from parts of the plane, to compressibility and altitude errors, and to variations due to gusts, as well as to possible errors in the indicating gage. In addition, these instruments indicate relative air speed only, and corrections are necessary to obtain absolute or ground speed.

4. Area Meters and Head-area Meters.—The **weir meter** is an open-channel or tank meter for liquids. The registering device is actuated according to the height of the surface of the liquid; hence both the head of the liquid and the area of the weir are involved (see Experiment 58 and Fig. 166).

Area meters for gas and air are of two types, the variable-orifice and the multiple-orifice meters. Figure 154 is a diagram of a typical variable-orifice area meter. Such a meter operates at a constant differential head, as determined by the weight of the movable unit, and it is self-adjusting for wide variations in the flow rate. Since area meters are essentially orifice meters they are subject to most of the advantages and disadvantages of fixed-orifice meters, but they have a much wider range of capacities. Their accuracy is usually within ±2 per cent.

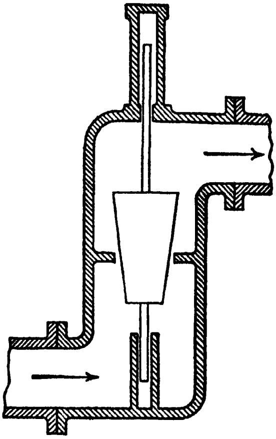

Fig. 154.—Variable-orifice area meter. (*Fluid Meters, A.S.M.E.*)

Several manufacturers offer gate type variable orifices, or multiple-orifice units. Manual adjustments of the area are thus possible, in order to change the capacity range, but these devices operate on a variable head, and the area is not changed by the action of the fluid as it is in a true area meter.

5. Thermal Meters.—There are several types of thermal meters. The **Thomas meter** measures the change in temperature of air or gas when a known amount of heat is added. It consists essentially of two electrical resistance grid thermometers with an electric heater between. A current regulator automatically maintains a certain temperature difference; hence the flow of air or gas may be read directly from a wattmeter or a watt-hour meter.

There are three forms of the heated thermometer anemometer, used for measuring the velocity of free air streams. The **Yaglau heated thermometer** is a mercury-in-glass thermometer with an electrical heater winding over the bulb. With a given impressed voltage, the difference between the heated thermometer reading and the reading of a similar but unheated thermometer placed beside it in the stream is a function of the air velocity. The range of the instrument is changed by changing the impressed voltage. The **heated thermocouple anemometer** is a similar instrument which uses a thermocouple as the sensitive element instead of a glass thermometer. On account of the small lag, only a single thermocouple is necessary, with alternate readings, heated and unheated. This instrument has the advantage of being distant reading. The **kata thermometer** is a large-bulb alcohol thermometer with a short indicating range, say 95 to 100°F. The thermometer is preheated to a temperature above the top graduation on the scale and then exposed in the air stream. The rate of cooling, timed with a stop watch, is a function of the air velocity. By a suitable calibration (see Experiment 57), the thermometer becomes an air-velocity meter. Several types of kata thermometers have been developed in England.

Another thermal instrument is the **hot-wire anemometer** which measures the air velocity over a heated wire in terms of the changes in its electrical resistance. In a typical hot-wire anemometer, a fine wire of platinum, tungsten, or nickel is connected into one of the arms in a bridge circuit, and a milliammeter indicates the unbalance in this circuit as the wire is heated or cooled. The milliammeter may be calibrated directly in feet per minute, since its deflection is a function of the cooling effect due to air velocity. The range and sensitiveness of the instrument may be changed by varying the heating current through the hot wire.

6. Theory of Head Meters.—Since the orifice, the nozzle, the venturi meter, and the pitot or impact tube are by far the most common flow meters in engineering work, a detailed examination of the theory and use of these devices will be made. Three of these meters, as shown in Fig. 155, are modifications of one and the same thing, *i.e.*, a restriction in the line of flow. As would be expected, such a restriction causes a difference of pressure to exist between the original section and the restricted section. This pressure difference varies with the flow, and this fact makes possible the use of the venturi, the nozzle, and the orifice as fluid-measuring devices.

The equations that give the relation between pressure difference and rate of flow are derived primarily from the law of conservation of energy. By equating the energy at a given upstream section to the energy at the throat, and taking into account that the mass rate of flow is the same for both sections, it is possible to express the rate of flow in terms of measurable quantities. These measurable quantities are the areas of the two sections, the properties of the fluid at the upstream section (pressure, temperature, density, etc.), and the fluid pressure at the downstream section.

Table XXII furnishes a reference list of the final equations in convenient form for practical use, and the derivation of these forms consists of three steps: (1) writing the general energy equation (Bernoulli's theorem) for the assumed conditions; (2) solving this equation for downstream or throat velocity; (3) expressing the rate of flow as a product of the velocity and the area, $Q = AV$.

The energy at any given section may exist in the form of internal energy (U), of gravitational potential energy (Z), or potential energy of pressure (P/ρ or $P\bar{V}$), or of kinetic energy ($WV^2/2g$). In going from a section of higher to one of lower potential energy (*i.e.*, downstream), mechanical energy may be converted into heat either by viscous friction or by turbulence. Heat transfer may also occur between the sections. In symbols this statement becomes

$$Z_1 + U_1 + P_1V_1 + \tfrac{1}{2}MV_1^2 \pm Q \pm \text{Work} = Z_2 + U_2 + P_2V_2 + \tfrac{1}{2}MV_2^2. \quad (51)$$

Since the same fluid is involved in both the upstream and the downstream sections, the symbol W (pound of fluid) is not usually used at all in the general energy equation, as it cancels out of all terms.

Units and Symbols.—In the summary given in Table XXII, and throughout this discussion, the symbols used are as given in the following. Note that consistent English units are used, *i.e.*, foot, pound, second, with the exceptions noted in the table.

A = measured area of pipe or section, square feet.

H = specific heat content (total heat or enthalpy per pound).

$P = h\rho$ = static pressure at pressure connection, pounds per square foot absolute (where h is the static head in feet of the fluid being measured).

TABLE XXII.—FLOW EQUATIONS AND THEIR APPLICATION

Flow quantities are given per second. For flow quantities per minute use $60Q$ or $60W$. See pages 269–271 for units and symbols.

Eq. No.	Common name	Theoretical equation (derivation and limitations, pages 253 to 257)	Coefficients and correction factors, page 271	Fluid	Devices	Allowable pressures
(52)	Hydraulic equation.	$Q = A_2\sqrt{2g(\Delta h)} = A_2\sqrt{\frac{2g}{\rho_1}(\Delta P)}$	C, M, Y	Liquids, gases vapors	Orifice, nozzle venturi, Pitot tube, impact tube	No limits for liquids. Gases: $\Delta P < 0.10P_1$
(53)	Heat-drop equation.	$W = 223.8A_2\rho_2\sqrt{\Delta H}$	C, M	Steam, other vapors	Nozzle, orifice, and venturi	$\frac{P_2}{P_1} >$ critical*
(53a)	Heat-drop equation.	$W = 223.8A_2\rho_2\sqrt{c_p(\Delta T)} = 8.02A_2\rho_2\sqrt{\frac{k}{k-1}\left(\frac{P_1}{\rho_1} - \frac{P_2}{\rho_2}\right)}$	C, M	Air, other gases	Nozzle, orifice, and venturi	$\frac{P_2}{P_1} >$ critical*
(54)	Perfect gas equation, complete form	$W = \frac{P_1A_2}{\sqrt{RT_1}}\left(\frac{P_2}{P_1}\right)^{\frac{1}{k}}\sqrt{2g\left(\frac{k}{k-1}\right)\left[\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}}}\right]}$	C	Gases and vapors	Nozzle, orifice, and venturi	$\frac{P_2}{P_1} >$ critical*
(55)	Perfect gas equation, no velocity of approach	$W = \frac{P_1A_2}{\sqrt{T_1}}\sqrt{\frac{2g}{R}\frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{1+k}{k}}\right]}$	C, M	Gases, vapors	Nozzle, orifice, and venturi	$\frac{P_2}{P_1} >$ critical*
(56)	Fliegener's equation	$W = \frac{0.53P_1A_2}{\sqrt{T_1}}$	C, M	Air	Nozzle	$\frac{P_2}{P_1} < 0.53$
(57)	Goudie's equation...	$W\dagger = 0.3A_2\sqrt{P_1\rho_1}$	C, M	Saturated steam	Nozzle	$\frac{P_2}{P_1} < 0.58$
		$W\dagger = 0.315A_2\sqrt{P_1\rho_1}$	C, M	Superheated steam	Nozzle	$\frac{P_2}{P_1} < 0.545$
(58)	Grashof's equation	$W\dagger = 0.0165A_2P_1^{0.97}$	C, M	Saturated steam	Nozzle	$\frac{P_2}{P_1} < 0.58$
(59)	Napier's rule........	$W\dagger = \frac{P_1A_2}{70}$		Steam	Nozzle	$\frac{P_2}{P_1} < 0.58$
(60)	Fliegener's equation	$W\dagger = 1.06\frac{A_2}{\sqrt{T_1}}\sqrt{P_2(P_1 - P_2)}$	M	Air	Rounded entrance nozzle	$\frac{\Delta P}{P_1} < 0.10$
(61)	Napier's equation...	$W\dagger = 0.0292A_2\sqrt{P_2(P_1 - P_2)}$	M	Steam initially dry saturated	Rounded entrance nozzle	$\frac{\Delta P}{P_1} < 0.10$

* May be used for $P_2/P_1 <$ critical by substituting P_t = critical pressure for P_2.

† Use pressure in pounds per square inch and area in square inches.

$Q = W/\rho$ = volume of fluid flowing, cubic feet per second.
T = absolute temperature of fluid at pressure connection, degrees Fahrenheit.
V = velocity (not volume) of fluid at pressure connection, feet per second.
W = weight of fluid flowing, pounds per second.
ρ = density of fluid at pressure connection, pounds per cubic foot (reciprocal of specific volume $\bar{V}$).

Subscripts.—Subscript 1 refers to the conduit and the fluid at the upstream pressure connection. Subscript 2 refers in the theoretical equation to the conduit and the fluid at the section of minimum area, but in actual use it may not be possible to measure the area, the pressure, temperature, etc., at the smallest section of the stream. In the latter case the subscript 2 refers to the smallest measurable area and to the conditions of the fluid at the actual pressure connection. The correction is then included in the coefficient of discharge (see page 272).

All the equations of Table XXII may be derived from the general energy equation [Eq. (51)] by the steps already indicated. It is assumed in all cases that there is no heat transfer ($Q = 0$), no external work in expansion or compression (Work = 0), and that the flow unit is in a horizontal position ($Z_1 = Z_2$). Equation (52) is derived for an incompressible fluid. Equation (53) is derived by substituting $H = U + (P/\rho 778)$ and for vapors the heat-content or enthalpy values may be taken from tables or charts. All the other equations are derived by the application of the laws of "perfect" or ideal gases (Table XX, page 235), using the constants for air and steam given in Table XIX, page 233.

7. Coefficients and Correction Factors for Head Meters.—Equations (52) to (58) are theoretically correct for the ideal fluids, and corrections to actual conditions are made by the coefficients Y, M, and C as indicated. **Values of Y** are given in Table XXIII, and this correction is applied only when using the hydraulic equation for gases and vapors.

The **meter constant M** is a correction for velocity of approach or upstream kinetic energy to be applied to those equations that otherwise neglect this factor, and it is calculated by Eq. (62):

$$M = \sqrt{\frac{r^2}{r^2 - 1}} = \sqrt{\frac{R^4}{R^4 - 1}} = \sqrt{\frac{1}{1 - \frac{1}{R^4}}}, \qquad (62)$$

where R = *diameter ratio* or ratio of upstream pipe diameter to throat diameter.
r = ratio of pipe *area* to throat area.

The **coefficient of discharge C** accounts for jet contraction and friction and must be selected for the particular conditions involved. Values of C are given in the following sections, applying separately to orifices, nozzles, and venturi meters (see Figs. 156, 157, 158, and 160). Equations (59), (60), and (61) already contain a coefficient C, but these equations are not so accurate as some of the others. Equation (59) is probably within 2 to 4

TABLE XXIII.—APPROXIMATE VALUES OF THE EXPANSION FACTOR Y

This correction for gas compressibility is to be used when applying the hydraulic equation [Eq. (52)] to the flow of air or gases through a venturi, flow nozzle, or orifice.

Ratio of pressures		Ratio of throat diameter to pipe diameter D_2/D_1					
$\frac{P_2}{P_1}$	$\frac{P_1 - P_2}{P_1}$	0.25	0.50	0.60	0.70	0.75	0.80
		For venturi meters and flow nozzles					
0.98	0.02	0.989	0.988	0.987	0.984	0.981	0.978
0.96	0.04	0.978	0.976	0.974	0.969	0.964	0.958
0.94	0.06	0.967	0.965	0.961	0.955	0.948	0.938
0.92	0.08	0.956	0.953	0.948	0.940	0.932	0.919
0.90	0.10	0.945	0.941	0.935	0.925	0.915	0.900
0.88	0.12	0.933	0.928	0.921	0.909	0.898	0.881
0.86	0.14	0.922	0.916	0.907	0.895	0.881	0.863
0.84	0.16	0.909	0.903	0.894	0.880	0.865	0.845
0.82	0.18	0.897	0.890	0.882	0.865	0.849	0.828
0.80	0.20	0.885	0.878	0.867	0.851	0.834	0.810
		For square-edged orifices					
0.98	0.02	0.994	0.994	0.994	0.993	0.993	0.992
0.96	0.04	0.986	0.986	0.986	0.985	0.985	0.984
0.94	0.06	0.980	0.980	0.979	0.979	0.977	0.976
0.92	0.08	0.974	0.974	0.973	0.972	0.970	0.969
0.90	0.10	0.970	0.969	0.968	0.964	0.962	0.961
0.88	0.12	0.964	0.963	0.962	0.959	0.956	0.953
0.86	0.14	0.958	0.956	0.954	0.952	0.948	0.946
0.84	0.16	0.952	0.950	0.948	0.944	0.940	0.937
0.82	0.18	0.947	0.944	0.942	0.937	0.933	0.929
0.80	0.20	0.941	0.938	0.935	0.930	0.925	0.920

NOTES: Based on density of the gas at the upstream pressure.

Values apply for gases having a specific heat ratio $k = 1.4$, but they may be used for gases for which $k = 1.3$ with a maximum error of less than 1.0 per cent. From "Fluid Meters," Part 1, A.S.M.E. Standard Code.

per cent for steam at moderate pressures, initially dry saturated. The error for Eq. (60) varies from less than 0.5 per cent when the ratio of the differential pressure to the absolute pressure is less than 15 per cent, to about 3 per cent when this ratio reaches 50 per cent.[1]

Corrections for steam quality and superheat may be applied to Eqs. (57) and (58). The numerical factor 0.315 in Eq. (57) should be reduced somewhat if only small or moderate superheats are involved. Equation (58) may be corrected as follows:

For quality, multiply by $1/\sqrt{Q}$.

For superheat, multiply by $1/(1 + 0.00065S)$, where Q is the quality expressed as a decimal and S is the degrees of superheat.

In ordinary metering practice *no* corrections are made for deviations from the ideal conditions due to heat transfer, gravity effect in a nonhorizontal unit, pulsating or unsteady flow, spiral or helical flow, supersaturation or two-phase velocities in a vapor, Joule-Thomson effect, or variation of specific heats in gases. For very accurate work, or for flow measurements at high temperatures a *correction for thermal expansion* of the primary element is required. This correction is about +0.5 per cent at 400° and 1.0 per cent at 700°.

8. Static Pressure Measurements.—When applying any one of the "head-meter" devices to the measurement of fluid flow, one common difficulty should always be kept in mind, *viz.*, the *difficulty of obtaining accurate static pressure determinations.* A great variety of shapes, sizes, and numbers of static holes have been investigated experimentally. According to "Fluid Meters, Part 1," A.S.M.E., the fundamental definition of static pressure is "the pressure that would act on and be indicated by a pressure gage moving along with the stream so as to be at rest or 'static' relatively to the fluid about it." Since such a measurement is impracticable, some substitute must be used, and the difficulty encountered is that either impact or suction is likely to be produced by the moving fluid at a "static" hole. The indicated pressure is then correspondingly higher or lower than the true static pressure, the latter error being the more common.

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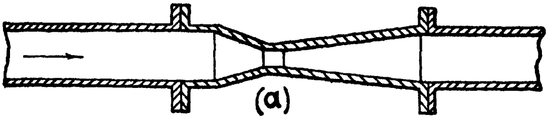
[1] For a more accurate modification of this equation, see S. A. Moss, "Measurement of the Flow of Air and Gas with Nozzles," *Trans. A.S.M.E.*, 1927, A.P.M. 3. Other papers by Dr. Moss in *Trans. A.S.M.E.*, 1928 and 1929, give valuable discussions on air and gas metering.

The two accepted methods for measuring static pressure are by means of a small hole in the side of the pipe, and by a static tube or a pitot tube inserted in the stream. For a pipe-wall connection the hole must be drilled perpendicular to the wall, and the engineer should personally check each connection to see that there are no projections or burrs on the inside pipe surface. The size and number of holes will depend on the size and smoothness of the pipe and the possibility of the holes becoming clogged with sediment. Holes 0.02 to $\frac{1}{16}$ in. diameter are probably the most accurate, if they can be kept open. A "piezometer ring" of multiple holes connected in parallel is sometimes used. The Fluid Meters Committee recommends a $\frac{1}{8}$- to $\frac{3}{16}$-in. hole for pipe sizes to $2\frac{1}{2}$ in., and a $\frac{1}{4}$- to $\frac{1}{2}$-in. hole for larger sizes to 16 in. The A.S.M.E. Code on Instruments and Apparatus (Pressure Measurements, Chap. II), recommends the use of tap-drill sizes for $\frac{1}{4}$-, $\frac{3}{8}$-, or $\frac{1}{2}$-in. pipe. The inside edge should be rounded to a radius of $\frac{1}{32}$ to $\frac{1}{16}$ in.

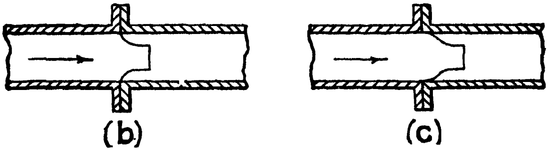
Static connections should be located as far as possible from elbows, valves, and fittings. Longitudinal straightening vanes should precede the connections if spiral or centrifugal effects are suspected.

9. The Thin-plate Orifice.—A square-edged, round hole in a thin metal plate ($\frac{1}{16}$ to $\frac{1}{8}$ in. thick) is the most widely used "primary device" for head meters. It is simple, inexpensive, easy to make, and easy to install between pipe flanges or at an intake or discharge opening. The upstream face of an orifice must have a 90-deg corner, with no visible burrs or rounding, and the cylindrical edge should not exceed 5 per cent of the orifice diameter. The plates must be flat and smooth and strong enough to resist bulging.

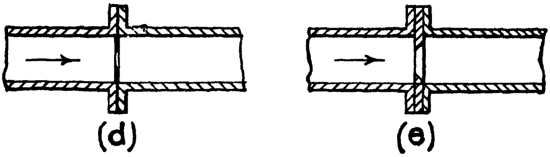
Since a thin-plate orifice does not guide the stream in natural flow lines, the *vena contracta* or smallest section of the jet does not occur at the orifice (see Fig. 155). Hence the **pressure connections** must be located very accurately, according to one of the three arrangements recognized by the A.S.M.E. Code: (1) *Radius taps,* with the upstream connection one pipe diameter from the upstream face, and the downstream connection one-half diameter from the downstream face. (2) *Vena contracta taps,* with the upstream connection one pipe diameter from the upstream face, and the downstream connection at the *vena contracta* (Fig. 156).



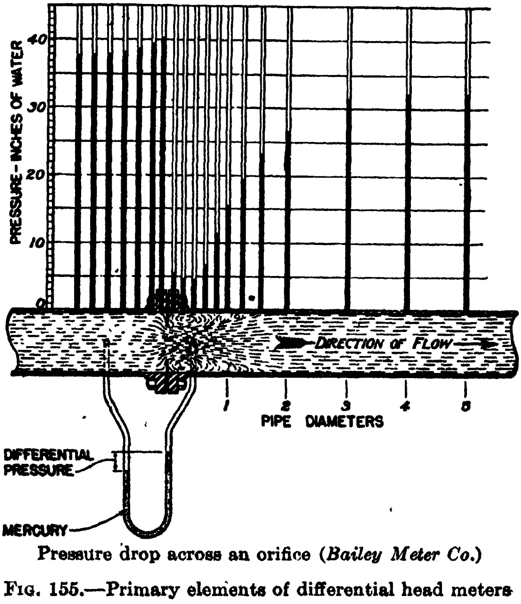
Venturi meter



Flow nozzles



Thin-plate or sharp-edge orifices



Pressure drop across an orifice (*Bailey Meter Co.*)

Fig. 155.—Primary elements of differential head meters

(3) *Flange taps*, with the center of each pressure connection 1 in. from the near face of the orifice plate. Two other arrangements of pressure connections are sometimes used, the *corner taps* at the orifice face (widely used in Europe), and the *pipe taps*, which are usually 2.5 diam upstream and 8 diam downstream. Pressure connections should be made as specified under "Static Pressure Measurements" page 273.

The **coefficient of discharge C** for thin-plate orifices is **approximately 0.60,** but this value may be in error as much as 5 per cent in some cases. Hence it is necessary to use graphs or tables to select the coefficient. For pipe orifices with either *vena contracta* taps or radius taps the values given in Fig. 156 are usually sufficiently accurate, but for greatest accuracy the coefficient should be selected in accordance with the Reynolds number (Figs. 157 and 158). Orifice coefficients are accurately known to 0.5 per cent or less, for all ordinary conditions. At low Reynolds numbers the orifice coefficient may change greatly.

Intake and discharge orifices, located at the end of a pipe instead of in a continuous run, are very convenient, especially for air measurement. In the absence of a standard table of coefficients for such orifices, it may be noted that $C = 0.60$ is probably correct within 1.5 per cent for Reynolds numbers above 100,000, for any diameter ratio up to $D_2/D_1 = 75$ per cent, and for either corner tap, flange tap, or in the case of intake orifices for a tap 0.4 pipe diameter downstream.

The over-all **pressure loss** due to a thin-plate orifice in a pipe line varies from about 40 to 90 per cent of the differential pressure, as shown in Fig. 156.

10. The Flow Nozzle.—The nozzle as used for fluid measurement usually consists of a smooth well-rounded approach or convergent portion and a parallel-throat section of appreciable length. The chief advantage of the nozzle is its nearly constant discharge coefficient. Typical flow nozzles are shown in Fig. 159, but adequate standards have not yet been established, and much less experimental work has been done on flow nozzles than on orifices.

For pressure connections located one diameter upstream and in the parallel-throat portion downstream, a **coefficient of discharge** $C = 0.985$ is probably within 1.5 per cent for diameter ratios of 20 to 70 per cent, in pipes 2 to 12 in. diameter. For

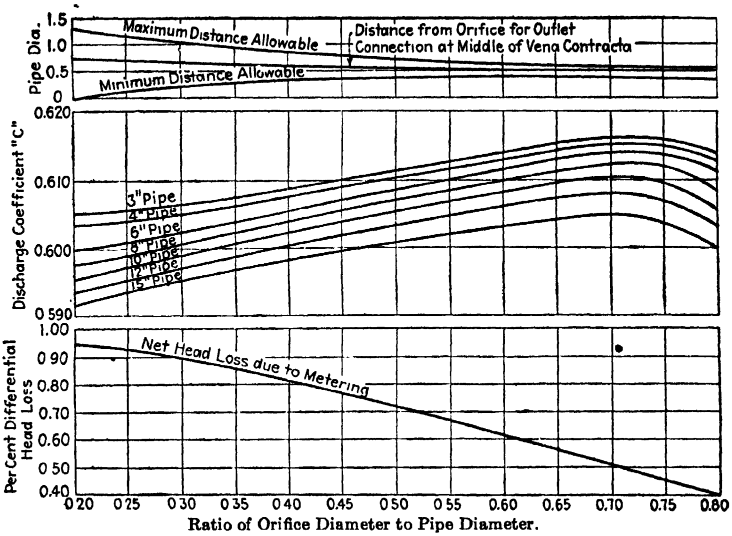


FIG. 156.—Orifice coefficients and characteristics with inlet connection at one pipe diameter from orifice and outlet connection at vena contracta for circular concentric orifices in pipes. (*From "Fluid Meters. Part 1," A.S.M.E.*)

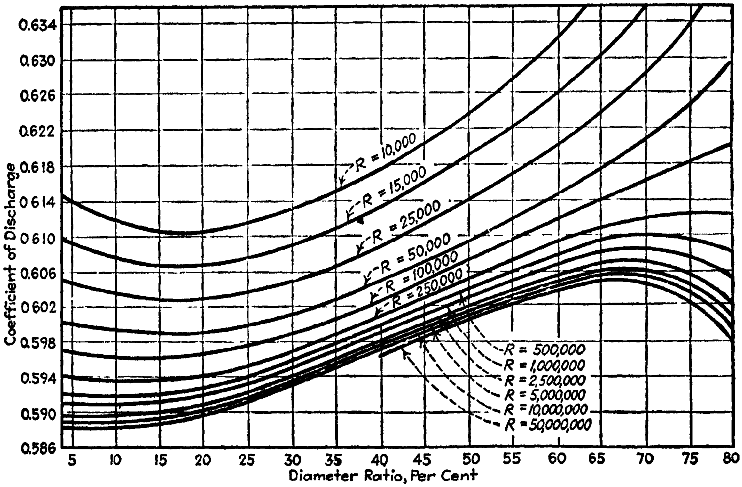
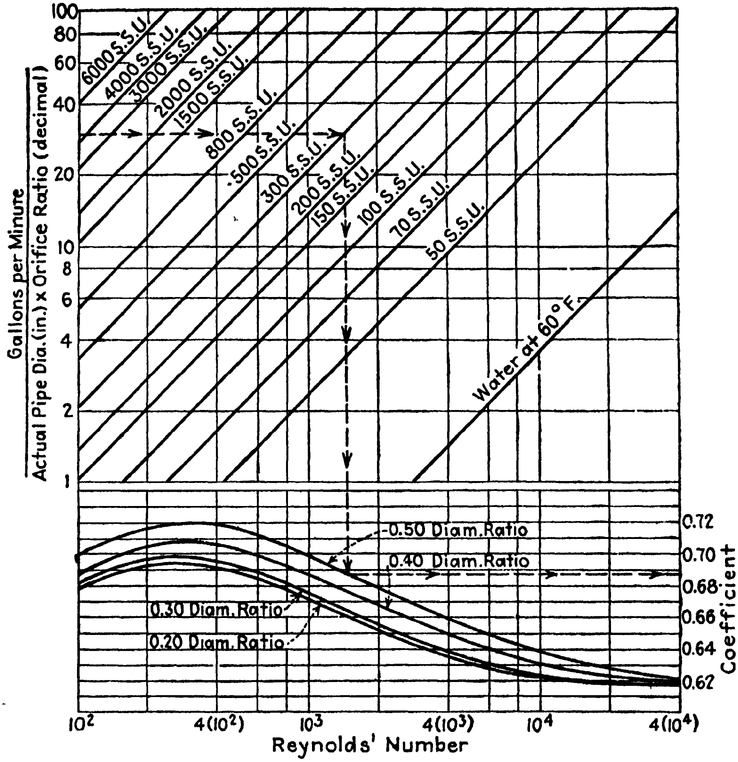


FIG. 157.—Variation of discharge coefficients for sharp-edge orifices (vena contracta connections) with Reynolds number. Tests made at the Ohio State University.



EXAMPLE: Assume a flow of 30 gal. per min. of 300 seconds Saybolt Oil through a 1" orifice in a 2" pipe. Proceed horizontally from the upper ordinate of $^{30}/_{2 \times \frac{1}{2}}$ to the 300 S.S.U. line, then drop vertically to the 0.50 diameter ratio line. The horizontal projection of this intercepted point is the resulting coefficient.

FIG. 158.—Orifice discharge coefficients for viscous liquids.

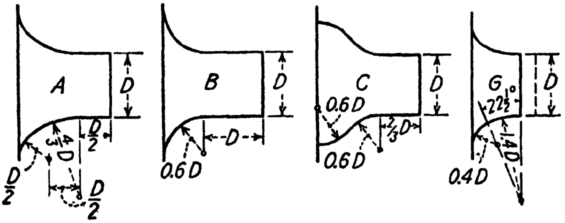


FIG. 159.—Proposed flow-nozzle standards. *A*, approximately as suggested by Bureau of Standards Research Paper 49, 1929. *B*, Compressed Air Society standard. *C*, Proposed by S. A. Moss. *G*, German (V.D.I.) "normal" nozzle; parallel portion may be added as shown by dotted lines. For all designs the pipe diameter should preferably be 2*D* or larger.

greater accuracy, the coefficient of discharge should be selected according to the Reynolds number (Table XXIV). For nozzles with free-open discharge to atmosphere (no downstream pressure tap) the coefficients are about 0.5 per cent lower.

TABLE XXIV.—FLOW-NOZZLE COEFFICIENTS

Variation of flow-nozzle coefficients with Reynolds number at throat. These coefficients apply to long-radius nozzles with pressure taps one diameter upstream and either in the parallel-throat portion downstream (not less than ⅜ in. from the end), or in the pipe wall ½ diameter downstream from nozzle inlet.

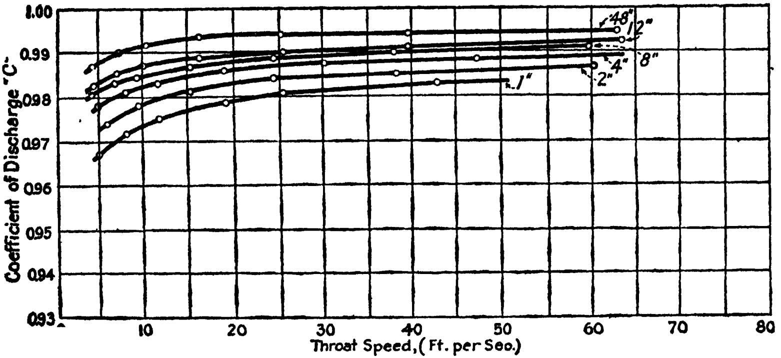
$\mathrm{Re} = DV\rho/\mu$	C
5,000	0.91
10,000	0.94
20,000	0.956
30,000	0.961
50,000	0.968
75,000	0.973
100,000	0.978
200,000	0.986
300,000	0.992

The over-all **pressure loss** due to the insertion of a nozzle is about the same as that for an orifice, because there is no diverging tube to give an orderly transformation from velocity pressure to static pressure.

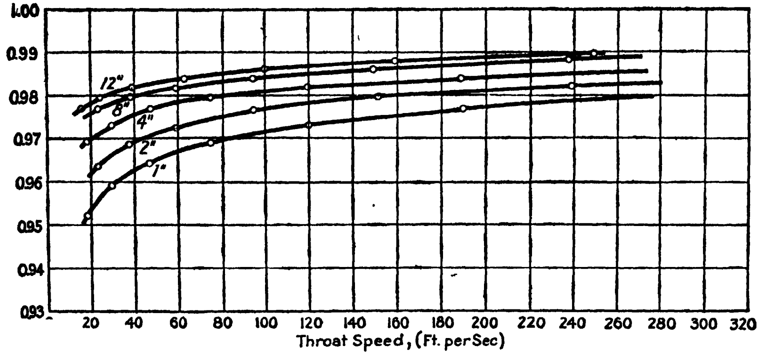
11. The Venturi Tube.—The form of the venturi tube (Fig. 155, page 275) is usually "a conical, nozzlelike reducer followed by a more gradual enlargement to the original size." The venturi principle has a great many applications, however, and the form of the venturi will depend upon its use.

The essential feature of the device is the static pressure difference between the entrance and the throat. The *absolute* static pressure at the throat will depend upon the proportions of the venturi and upon the absolute initial pressure. The throat pressure may under certain circumstances be well below atmospheric; in fact it may be only a small fraction of 1 lb abs, as in the application of the venturi construction to the jet vacuum pumps used on large surface condensers (see Chap. VIII, page 308).

When the venturi is used for fluid measurement, the throat section is comparatively large and the "venturi tube" usually takes the shape of two truncated cones joined at their smallest diameters by a short, cylindrical throat piece and preceded by a cylindrical entrance section the same diameter as the pipe (see Fig. 155). The entrance and throat sections in particular must



Venturi Discharge Coefficients for Water at 60 Deg. Fahr.
C is Defined from $V = CM\sqrt{2gh}$



Venturi Discharge Coefficients for Air at 60 Deg. Fahr. and one Atmosphere
C is Defined from $V = CM\sqrt{2gh}$

FIG. 160.—Venturi discharge coefficients for various throat diameters. ("*Fluid Meters, Part* 1," *A.S.M.E.*)

be finished smooth and true and their diameters accurately determined, as the **pressure connections** are made at these points. The total angle of the entrance cone is about 21 deg and that of the exit cone 5 to 7 deg, the diameter of the throat being usually between one-half and one-fourth of the entrance diameter. Sometimes a rounded entrance is used, and the diverging cone may have a wider angle without affecting the coefficient of discharge.

Very small venturis, say for pipe lines less than 2 in. in diameter, are commonly made of brass or bronze and smoothly finished all over the inside to reduce friction. Larger venturis are usually of cast iron, the throat and sometimes the straight entrance portion being lined with brass or bronze and machined to a smooth finish. Very large venturis, up to 20-ft pipe diameter have been made of smooth-surfaced concrete, the throat only being of machined bronze. Wood staves and steel plate have also been used for the cones of large venturis. The two large venturis at the Conowingo hydroelectric station are approximately 42 by 24 ft.

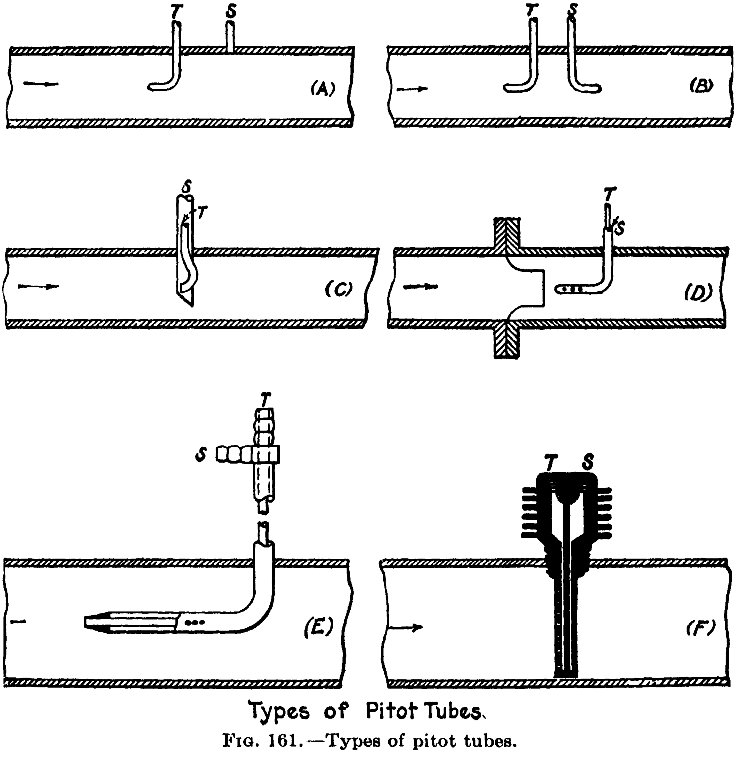


FIG. 161.—Types of pitot tubes.

The **coefficient of discharge** for a venturi tube depends on the Reynolds number, but it also varies somewhat with diameter ratio, and with pipe size. Approximate coefficients for water and air are given in Fig. 160.

One of the advantages of the venturi meter is that the **loss of head** due to its installation in the pipe line is small, the over-all loss usually being only 10 to 20 per cent of the differential head.

12. Pitot Tubes and Impact Tubes.—The pitot tube differs from the other head meters already considered in that it does not depend for its indications upon a difference in static pressure at two sections in the stream. No constriction is placed in the stream, the differential pressure being obtained by measuring both the static pressure and the total pressure at the same location.[1]

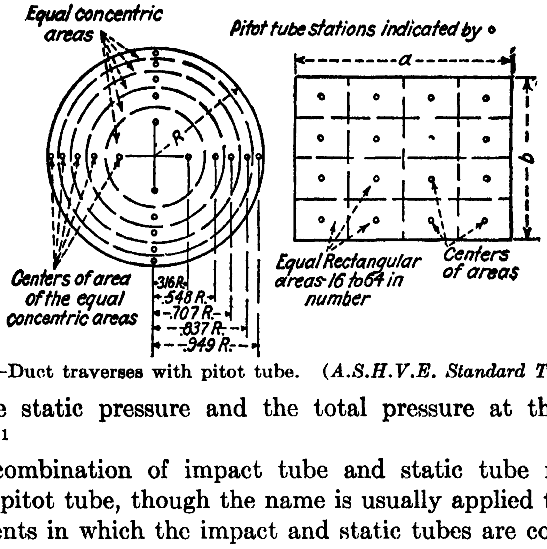

Equal concentric areas
Pitot tube stations indicated by ○
a
b
R
Centers of area of the equal concentric areas
.316R
.548R
.707R
.837R
.949R
Equal Rectangular areas-16 to 64 in number
Centers of areas

FIG. 162.—Duct traverses with pitot tube. (*A.S.H.V.E. Standard Test Code.*)

Any combination of impact tube and static tube may be called a pitot tube, though the name is usually applied to those instruments in which the impact and static tubes are combined into a single device. Figure 161 illustrates several types, and the static pressure and total pressure connections are indicated for each type.

The simple hydraulic formula [Eq. (52), page 269)] is applicable to the pitot tube, unless very high velocities of compressible fluids are to be measured. In the latter case, for velocities below that of sound, the adiabatic formula for velocity [similar to Eq. (55), page 269], should be used. The error of the hydraulic equation for air at atmospheric pressure reaches 1 per cent at a velocity of about 220 ft per sec (150 mph). Hence the U. S. Army and Navy have adopted the adiabatic formula to be used for the calibration of pitot-tube airplane-speed meters.

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[1] See Pressure Measurements (Chap. II, p. 12), for definitions of and methods for measuring the various kinds of pressure.

The **coefficient is unity** for the standard forms of pitot tubes (Fig. 161*A* and *E*, and Fig. 164). When the static tubes do not measure true static pressure, as in the case of those shown at *B*, *C*, and *F* in Fig. 161, a special calibration is necessary.

In spite of its high accuracy on individual readings, the obtaining of accurate flow measurements with a pitot tube is likely to be a laborious process. Since the velocity calculated from a single set of readings is the velocity at one location only (*i.e.*, at the location of the impact opening when the readings were taken), it becomes necessary to take several sets of readings in order to obtain the *average* velocity across a given section. In large ducts this **traverse** will require 20 or more sets of readings. These readings are taken on two diameters, at the center of areas of each concentric area, as shown in Fig. 162. Layout dimensions for pitot-tube locations are given in Table XXV. Connections for pitot tube are as shown in Fig. 163.

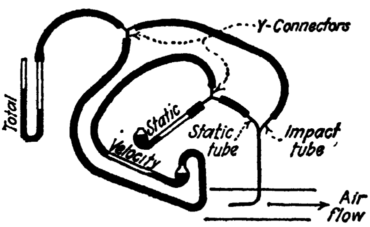

Fig. 163.—Diagram of connections for pitot tube and manometers.

Table XXV.—Layout Measurements for Pitot-tube Traverse

Number of equal areas	Total number of readings	Distances from center of pipe to point of reading in per cent of pipe diameter				
3	12	20.4	35.3	45.5		
4	16	17.7	30.5	39.4	46.6	
5	20	15.5	27.2	35.3	41.7	47.4

The equivalent velocity for each velocity pressure reading is to be calculated, and the average of the calculated *velocities* is taken as the average velocity of the fluid in the pipe. For approximate work, the velocity *pressures* may be averaged and the average velocity calculated from the result.

The use of the pitot tube in a fixed position in a pipe is not recommended, because the velocity pattern depends upon both the upstream conditions and the Reynolds number. Average velocity is sometimes assumed to exist at a distance of about 0.15 diameter from the pipe wall. Another rough approximation is that the average velocity is 83 per cent of the velocity in the center of the pipe.

Pitot tubes and impact tubes have been used in pipes of all sizes and with velocities up to 25,000 ft per min. For investigating stream flow in carburetor tubes and like devices, impact tubes as small as $\frac{1}{32}$ in. outside diameter have been successfully used. The pitot tube is also the standard primary device for speed meters on airplanes.

The form of pitot tube shown in Fig. 164 has been adopted as the standard for use in tests of fans and blowers. The accuracy of this tube is less affected by slight errors in positioning, than was the case for the former design (Fig. 161*E*).

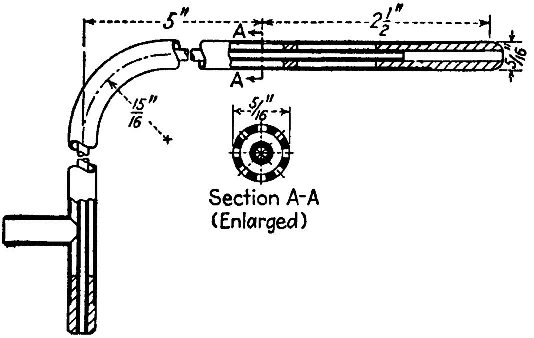

FIG. 164.—Standard pitot-static tube for use in fan tests. (*A.S.H.V.E. Code.*)

Velocities corresponding to impact pressures of only a few thousandths of an inch of water may be successfully measured by using a suitable micromanometer. With very small tubes and low velocities, a viscosity correction may be necessary.

The impact tube *G* of Fig. 170 is sometimes used independently, either to measure the velocity of a fluid issuing from a nozzle or as a substitute for the static tube.

As a substitute for the pitot tube, the use of an impact tube in the discharge jet from a properly shaped flow nozzle has several advantages, particularly in the testing of air compressors and blowers discharging to atmosphere. The necessity for a traverse is in the case of an impact tube practically obviated, because the flow nozzle so straightens the stream lines that the velocity of the issuing jet is practically the same at all places, with the exception of the very edges.

EXPERIMENT 57. Calibration and Use of an Anemometer

I. PREFACE

Anemometers, or air-velocity meters, are usually mechanical vane type instruments, although several thermal anemometers have recently been developed. The mechanical instruments are of two kinds: (1) The current meters (Fig. 165) in which a propeller vane wheel or cup-rotor operates in continuous rotation.

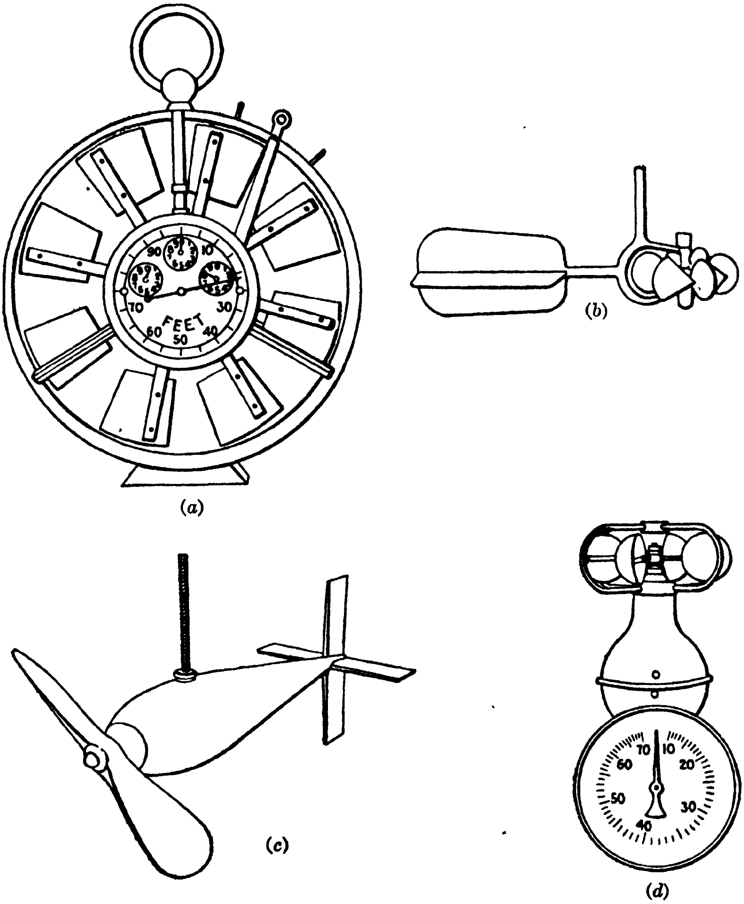

FIG. 165.—Types of current meters.

(2) The bridled-vane type, in which the deflection of a vane or wheel is resisted by a spring, thus giving a direct-reading indicating instrument (such as the velometer).

Among the anemometers of the thermal type are the kata thermometer, the heated-thermometer anemometer, the heated-thermocouple anemometer and the hot-wire anemometer. These instruments are briefly described under "Thermal Meters," page 267. Air-speed instruments for airplanes are usually pitot tubes rather than mechanical or thermal anemometers (see page 266).

Each type of air-velocity instrument is favored for certain uses. The Weather Bureau uses the cup type for wind measurements. The velometer is widely used for quick checking of grille velocities in ventilation and air conditioning. The kata thermometer has been used mainly in room-comfort studies. It is the least expensive of all the instruments. The heated-thermocouple and the hot-wire instruments are distant reading. These instruments and the heated thermometer have been used chiefly in research work, for measuring velocities below 100 ft per min. The hot-wire anemometer has also been widely used in aerodynamic studies. Its low thermal inertia makes it adapted for use with the oscillograph, measuring rapid fluctuations of velocity and flow.

The rotating-vane instrument (Fig. 165*a*) is by far the most common anemometer. It is used mainly for air measurements at inlets and outlets, and in large ducts. Most rotating anemometers are 3 to 6 in. in diameter, and are adapted to air velocities from 100 to 2,000 ft per min. When the anemometer is accurately calibrated and used, it deserves to be classed as an accurate air meter, as it will duplicate its own readings consistently within less than 1 per cent.

II. PREPARATIONS

1. Apparatus.—There are three satisfactory methods for anemometer calibration, using (*a*) a rotating arm, (*b*) a suitable wind tunnel, and (*c*) a flow meter. The first two are suitable for *velocity* measurements, but the third is applicable only when the desired result is air *quantity*. For greatest accuracy the rotating arm should be at least 10 ft long, although consistent results may be obtained with a 30-in. arm (error about +5 per cent). For the wind-tunnel method, a draw-through tunnel with a very large bellmouth is desirable, although a blow-through tunnel is satisfactory if the entrance nozzle has a large reduction ratio. An anemometer should never be calibrated in an ordinary fan duct, as the velocity pattern is too uncertain. The wind-tunnel method is satisfactory only for the higher velocities, where a pitot tube with calibrated micromanometer may be used as the standard. The flow-meter method of calibration involves the accurate metering of all the air that passes the anemometer-traverse section. This method is used in cases where an anemometer and test pipe are to be calibrated as a unit.

III. INSTRUCTIONS

1. Procedure.—The rotating-arm calibration is the simplest and most widely used. Bolt the anemometer *securely* to the end of the arm, and arrange the starting trigger or lever so that it may be manipulated while the arm is rotating. One convenient method is to use a rubber band and string, running the string along the arm, and out through a hole at the axis, attaching the rubber band in such a way that the anemometer dials will operate at all times except when the string is pulled tight. Rotate the arm at various constant speeds as required, for a time interval of at least 1 min (or to the nearest full revolution of the arm) at each speed. Use five or more speeds, approximately equally spaced over the range to be covered (usually about 150 to 1,000 fpm). Form 57 is a suggested tabulation. **While the calibration is in progress, plot the results as a straight-line graph** of actual velocity in feet per minute (on the ordinates) against indicated velocity by instrument. Repeat any points that are obviously in error.

After calibration, use the anemometer for air measurements at an intake or discharge opening, as directed by the instructor. Divide the rectangular opening approximately into squares the size of the anemometer, making a grid of string or fine wire. A traverse consists of continuous timed reading of the anemometer dials when the instrument is held successively in the center of each square (about 10 sec in each position). Make traverses in triplicate for accurate check. Use another type of anemometer for the same measurement, if available.

2. Results.—The report should include a straight-line calibration graph as described, and a correction curve giving the correction, plus or minus in feet per minute as ordinates, against the anemometer reading in feet per minute.

Form 57.—Calibration of Anemometer

Description	Constants
Type..........................	Length of rotating arm.............ft.
Maker	Travel of one turn.................ft.
Laboratory No.................	Turns for 100 ft. travel...............
Range	

Experimental log

Time	Number of turns	Distance, ft.	Anemometer readings			Velocity ft. per min.		Correction + or −
			Start	Stop	Indicated distance	Actual	Indicated	

Date .... Observers:.

Give capacity of the equipment tested in cubic feet per minute, using "application factors" as instructed in the next section, "Notes" paragraph 6. Discuss accuracy of instruments, calibration methods, and final results.

IV. NOTES AND PRECAUTIONS

1. Do not touch or let anything touch the fan wheel. Should the blades become even slightly distorted, the accuracy of the instrument will be

2. Handle the instrument as you would a good watch. Its delicate jewel bearings are easily damaged.

3. Never place the anemometer where it will be subjected to air velocities greater than the maximum for which it was designed (2,000 ft per min is the maximum for most instruments).

4. Never use the anemometer in a current of air or gases at a temperature above 200°F.

5. It is important that the axis of the fan wheel be parallel to the air current and that the fan be operating at full speed before the connecting lever is engaged.

6. **Application factors** must be used for the determination of true air volume from average velocity by anemometer, according to the equation

$$\text{True volume} = \begin{pmatrix}\text{application}\\ \text{factor}\end{pmatrix} \times \begin{pmatrix}\text{average velocity}\\ \text{by anemometer}\end{pmatrix} \times \begin{pmatrix}\text{area}\\ \text{designated}\end{pmatrix}.$$

The "area designated" is the total area traversed in the case of all air intakes or in the case of air-discharge openings without grilles. The designated area for an air-discharge grille is the arithmetical average between the total area traversed and the free-open area between bars. *"Application factors" for general use are: For air intakes,* 0.85, *and for air-discharge openings* 1.03. These factors apply for Biram-pattern anemometers, 3 to 6 in. diameter, used for equal-time traverses over areas up to 500 sq in., and average velocities from 400 to 1,500 fpm. They apply to wall type or flanged intakes, but *not* to open-end sheet-metal ducts without flanges. They apply to a discharge opening at the end of a straight duct, but *not* to discharge openings in the sides of large ducts or plenum chambers. For air velocities below 400 fpm or for openings larger than 500 sq in., correction factors must be used (see "Air Flow Measurements at Intake and Discharge Openings and Grilles," Tuve and Wright, *A.S.H.V.E. Journal*, August, 1940). Accurate air-volume measurements cannot be made by anemometer when the air is discharged directly from an elbow, a stack head, or a fan outlet, unless special calibrations are made to determine the application factors.

EXPERIMENT 58. Calibration of a Weir

I. PREFACE

A commercial V-notch weir meter having a capacity of 1,000,000 lb of water per hour was recently sold to a large electric central station under a guarantee of accuracy to ½ per cent over its entire range, whether using full 90-deg weir plates or fractions thereof. Such a guarantee is very unusual for any type of commercial flow meter, and it well illustrates the high place this type of meter holds in the opinion of many engineers. In the station in which the above mentioned meter is installed, it is used as a primary standard against which various other flow meters are calibrated.

The chief advantage of the V-notch or triangular weir over the other types lies in the comparatively large change in head produced by a small change in the rate of flow. Thus, greater accuracy is secured, whether the head readings are taken by hook gage or by the mechanical recording and integrating devices of the commercial meters. Moreover, by substituting the fractional-size weirs the meter obtains the advantage of a very wide range, and the same flow formula and hence the same mechanical integrating devices are applicable over the entire range.

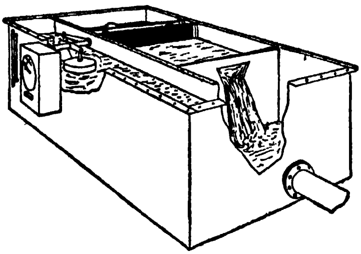

FIG. 166.—Weir meter with float-actuated indicator and recorder. (*Bailey Meter Co.*)

One disadvantage of this meter is that open-channel flow is necessary and a special meter tank or flow box must be used (see Fig. 166). Such a tank is often built integral with a feed-water heater. The conditions of use of the meter are limited and the cost increased, compared with a meter which may be installed in a pipe line.

The angle of the notch or apex may have any convenient value, but the 90-deg notch is the most common. The 60-deg notch is sometimes used for the measurement of smaller quantities, but a more convenient method is to make the area of the smaller weirs even fractions of the area of a 90-deg V-notch weir. Thus a "half-notch" weir has an angle of 53 deg 8 min, and a "quarter-notch" weir has an angle of 27 deg.

The weir equation is derived from the fundamental equation for the flow of noncompressible fluids [Eq. (52), page 269, see any textbook on hydraulics for derivation], and for the triangular weir the equation becomes

$$Q = C\sqrt{h^5}, \qquad (63)$$

where Q = volume of liquid flowing, cubic feet per minute.
h = head over the weir, inches, or distance of water level above the apex, usually from 3 to 24 in.

C = coefficient of discharge, very close to 0.3 for a 90-deg notch and to 0.15 for a "half-notch."[1] (This coefficient also involves a conversion factor.)

The weir is a combination head meter and area meter and is classified as a "head-area" type in the A.S.M.E. classification (see page 267).

In this experiment a V-notch weir is to be calibrated by determining the coefficient C to be applied in Eq. (63) for various rates of flow corresponding to certain values of the head above the apex of the notch. The head is to be measured by using either a hook gage or an inclined-tube manometer.

II. INSTRUCTIONS

Determine the zero level of the hook gage by laying a thin straightedge across from the apex of the notch to the top of the

Form 58.—Calibration of Weir

Form and dimensions of notch ....

Number of run	1	2	3	4	5
Duration of run, min....................					
Hook gage, average reading, ft.					
Hook gage, zero reading, ft....					
Head on weir, in........................					
Water over weir:					
Scale or tank reading, start............					
Scale or tank reading, stop.............					
Net water, lb........................					
Temperature of water, deg. fahr........					
Total, cu. ft.........................					
Cu. ft. per min......................					
Velocity of approach, ft. per sec..........					
Average coefficient of discharge...........					

Date................ Observers:......................................

....

....

---

[1] For curves giving discharge coefficients for full 90-deg, half-, quarter-, and eighth-notch weirs, see D. R. Yarnall, "Accuracy of the V-notch Weir Method of Measurement," *Mech. Eng.*, January, 1928. General weir equations and an alignment chart are given by Tarrant, *Trans. A.S.M.E.*, 1928, HYD50-8.

hook and adjusting the hook according to the indications of a spirit level placed upon the straightedge. If this method is inconvenient, the weir notch may be blocked off and a hook-gage reading made for a water level of known height above the apex of the notch. For accurately determining this height, a carefully measured gage block with a pointed top and a knife-edge bottom should be fixed in the notch and the water level adjusted as in using a hook gage (so that the point just penetrates the water surface).

The length of run and the heads to be used should be determined according to the size of the weir and the weighing or measuring tanks available. Make duplicate runs at not less than five heads, equally spaced over the available range, see Form 58. Calculate the coefficient of discharge from Eq. (63) and plot the results. Also plot a curve of water quantity against head on logarithmic paper, and determine the equation therefrom.

EXPERIMENT 59. Steam-nozzle Characteristics

I. PREFACE

Various methods have been used with more or less success for the study of flow characteristics in turbine nozzles: (1) impact of jet against a flat plate; (2) reaction of jet, measured by suspending the nozzle in a balance; (3) exploring tube for measuring static pressures; (4) exploring tube for measuring impact pressures; (5) photographs of the jet, usually with glass nozzles; (6) flow

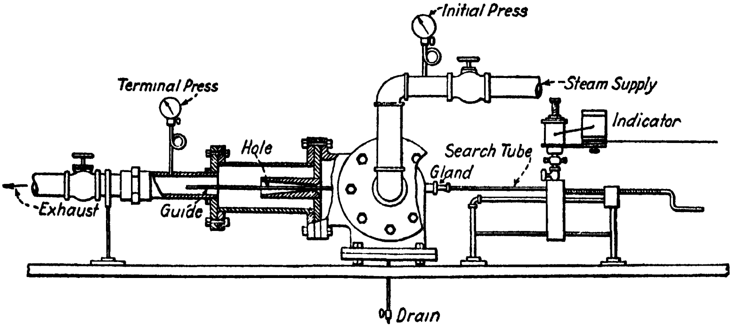

FIG. 167.—Steam-nozzle search apparatus.

patterns in plastic material (such as graphite-oil paste smeared inside the nozzle); (7) thermocouple traverses in the jet.

The exploring-tube method has been perhaps more widely used than any of the others, and in this experiment some of the outstanding characteristics of nozzle flow will be demonstrated by this method, using a static tube. For discussion of the other methods the references (page 294) should be consulted.

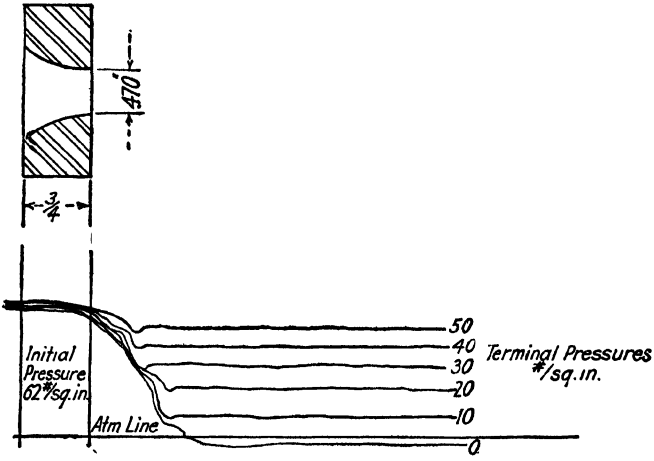

Fig. 168.—Rounded-entrance nozzle and pressure card.

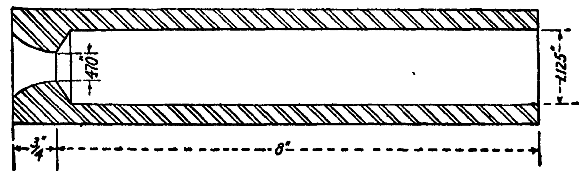

Fig. 169.—Types of steam nozzles.

II. INSTRUCTIONS

1. Apparatus.—The apparatus (Fig. 167) consists of a ⅛-in. brass tube closed at the extreme end and having a small hole drilled in one side, as shown. Through this hole the steam pressure is built up in the searching tube and transmitted to the indicator cylinder. The screw device permits translation of the indicator and searching tube so as to move the pressure hole through the entire length of the nozzle or orifice.

2. Procedure.—The position of the searching hole relative to the nozzle under test should be carefully ascertained at the start; note this position on the indicator card.

With maximum pressure on the entrance side of the nozzle or orifice, vary the back pressure by uniform increments, as illustrated in Fig. 168. Secure cards for each orifice and nozzle assigned (see Figs. 168 and 169).

3. Report.—Discuss in full the pressure curves, in regard to overexpansion, underexpansion, vibration, etc.

III. REFERENCES

1. STODOLA: "Steam and Gas Turbines," Vol. I, pp. 70–107.

2. WARREN and KEENAN: "Testing Steam Turbine Nozzles by the Reaction Method," *Trans. A.S.M.E.*, 1926, p. 33, and *Power*, Feb. 16, 1926.

3. "Turbine Nozzle Loss Analyzed by Air-flow Measurement," *Power*, Dec. 2, 1924.

4. Report of Steam Nozzle Research Committee, *Proc. Inst. Mech. Eng.* (England); abstracted in *Mech. Eng.*, March, 1924, p. 148; February, 1926, p. 171; October, 1928, p. 707.

EXPERIMENT 60. Orifices and Nozzles for Steam Measurement

I. PREFACE

In most cases calling for the measurement of steam, it is not convenient to condense the steam and weigh the condensate. In such cases, if the steam flow is free from pulsations, the orifice or nozzle is usually the most convenient measuring device. Most commercial steam meters use one of these devices (see Experiment 61, Flowmeters, page 296).

The type of apparatus to be used will depend largely on the steam-pressure range. The commercial meters use a comparatively large orifice and a small pressure drop. The present experiment will serve to illustrate the methods for measuring a small amount of steam with a comparatively large pressure drop across the measuring device. Calculated steam quantities determined by formulas will be compared with actual quantities obtained by condensing the steam and weighing.

II. PREPARATIONS

Apparatus.—The apparatus as set up in the laboratory consists of a steam-pipe system having a flange conveniently located for the insertion of an orifice or nozzle plate. Valves for controlling the steam flow are located on either side of the orifice flange. Between the valves and the flange are located gage connections, thermometer wells, and calorimeter connections. A rounded-entrance nozzle plate and a sharp-edge orifice plate are provided, the former having an additional gage connection for the measurement of throat pressure. The apparatus discharges to a surface condenser and from it to weighing tanks.

To prepare the apparatus for use, the rounded-entrance nozzle should be inserted in the flange and calibrated gages and thermometers of suitable range (usually 100-lb gages and 400° thermometers) should be installed. A calorimeter is to be installed between the upstream valve and the nozzle plate.

To warm up the apparatus, the discharge and drain valves should be opened wide and the steam-supply valve opened slightly. Drain valves must be closed before runs are started.

III. INSTRUCTIONS

1. Procedure.—Make five 20-min runs as follows: Select an upstream pressure somewhat below the steam-line pressure (say 60-lb gage) and make the first three runs with this initial pressure and back pressures as follows: (1) P_2 well below the critical pressure (see page 259); (2) P_2, 5 to 10 lb above the critical pressure; (3) P_2, 5 to 10 lb below the initial pressure; (4) use the same back pressure as was used in the first run, but lower the initial pressure at least 10 lb; (5) duplicate the conditions of the fourth run, but substitute the sharp-edge orifice for the rounded-entrance nozzle.

To make a run, set the pressures as closely as possible, and, after allowing 5 min for the establishment of constant conditions in the condenser, take readings at 5-min intervals **until four successive steam weighings check.** Use this interval as the 20-min run, neglecting other readings. There will be no difficulty in making the readings check if the steam-gage hands are maintained **exactly** at their predetermined settings.

2. Results and Calculations.—Results are to be calculated by all the available steam-flow equations (see Table XXII, page 269) and the results discussed with respect to both the accuracy of the experimental work and the accuracy of the equations and coefficients. Discharge coefficients should be selected from the values given in the discussion accompanying Table XXII.

EXPERIMENT 61. Flowmeters

I. OBJECT

The object of this experiment is to make a comparative experimental study of the simple differential metering devices such as form the bases of the various commercial meters employed in the measurement of air, steam, gases, water, oil, and other fluids.

II. PREFACE

A problem encountered very frequently by the mechanical engineer is that of measuring the quantity of a liquid or gas flowing in a pipe. The field of usefulness of the common domestic type water and gas meters has been rather limited, and it is only within the last few years that the present type of differential flowmeter has become commercial. The fact that reports of flowmeter investigations to the extent of several thousand pages have been published within the last 15 years indicates the live interest which is being shown in the problem (see bibliography in "Fluid Meters, Part 1," A.S.M.E.).

The flowmeter has a large variety of applications for measuring gases and liquids of all kinds. A few of the most common uses in the mechanical engineering field are as follows:

1. Steam Meters.—Boiler output; departmental steam consumption in industrial plants; steam used in heating, drying, and processing; steam consumption of main and auxiliary power units; steam used by soot blowers, ash conveyors, and other steam-jet appliances.

2. Air and Gas Meters.—Tests of compressors, blowers, fans, vacuum pumps, and gas boosters; air required for air tools, hoists, foundry equipment, oil and gas burners, conveyors, etc; metering of natural or manufactured gas for sale or purchase.

3. Meters for Liquids.—Hot or cold water in large quantities for boiler feed, process work, etc. Meters for oils of all kinds (except high-viscosity oils) especially when handled in large quantities. Solutions and liquors in the chemical engineering industries.

III. PREPARATIONS

Figure 170 is a sketch of the Bacharach demonstration-test outfit for comparative studies of standard head meters by placing them in series on a low-pressure air line. The apparatus will be found already set up in the laboratory essentially as shown in the sketch, but with the addition of a valve at the compressor (or blower) for further control of air volume and pressure. Each differential device is provided with its separate differential manometer. An additional manometer is also provided, which may be connected to the upstream side of one metering device at a time, for determining the static pressure at that device. The connections to all manometers should be checked and the zero levels adjusted and recorded before the blower is started.

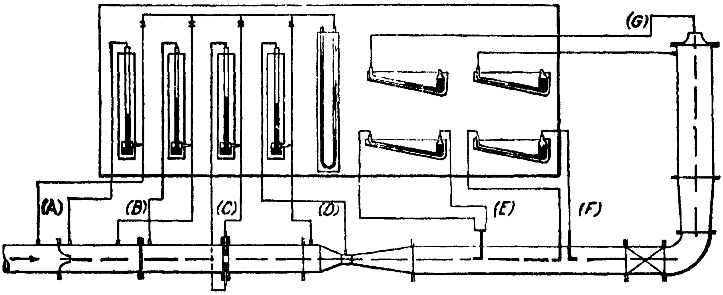

Fig. 170.—Flowmeter-test outfit.

Two 120° (or 220°) thermometers, a revolution counter, and data boards should be obtained at the instrument room.

IV. INSTRUCTIONS

1. Adjustments.—For maximum volume (and hence maximum velocity pressure), both of the large valves should be wide open.

For higher static pressures, the valve at the discharge end should be partly closed.

For low static pressures and low volumes, the valve at the blower should be partly closed.

2. Readings.—A set of readings consists of a velocity pressure and a static pressure reading for each device, together with readings of dry- and wet-bulb thermometers and of revolutions per minute of the blower.

Before reading the pitot gage, the pitot tube should be placed in the center of the pipe. The impact tube should also stand in the center of the jet.

The zero of each gage should be checked after completing the readings.

After a set of readings has been obtained at each of the various valve settings designated by the instructor, an investigation should be made of the effect of changing the location of the pressure taps or connections. With a fixed valve setting, a series of readings should be made on the thin-plate orifice, using pressure taps at various distances upstream and downstream from the orifice plate. These readings should be discussed and also shown graphically in the report, plotting static pressure on the ordinates against location of the pressure taps on the abscissas (Fig. 155).

3. Results and Calculations.—Dimensions of the various metering devices may be obtained from the instructor. All air quantities are to be expressed in cubic feet of air per minute at the pressure and temperature existing on the upstream side of the metering element; also in cubic feet per minute under standard conditions of 68°F and 14.7 lb abs, and in pounds per minute. Discuss agreement or disagreement of results. *Plot curves of differential pressure against rate of air flow in pounds per minute for each metering device.*

In order to save time, the following detailed instructions for calculation are suggested:

a. Flow Nozzle.—Read discussion page 276. When the differential pressure is 1 per cent or less of the upstream absolute pressure, use Eq. (52), page 269, correcting for velocity of approach by Eq. (62), page 271. When the differential pressure is more than 1 per cent of the upstream absolute pressure, use Eq. (60).

b. Thin-plate Orifice.—Read discussion page 274. Use Eq. (52), correcting for velocity of approach by Eq. (62) and for gas compressibility by a factor from Table XXIII, page 272. Obtain coefficient of discharge from Fig. 156, page 277. Check location of pressure connections to make sure that these correspond with those specified in Fig. 156. If other locations of the pressure taps are used, consult "Fluid Meters, Part 1," A.S.M.E., for coefficients.

c. Thick-plate Orifice or Short Tube.—This device is not well standardized and is used here largely for illustrative purposes. Use Eq. (52), correcting for velocity of approach by Eq. (62) and obtaining Q from results of calculations on venturi measurements, finally solving for C. In other words, the result sought in this case is the coefficient of discharge for the short tube.

d. Venturi, Type 2.21.—Read discussion page 279. The results obtained by using the venturi will be considered correct and the error of the results with other devices will be calculated accordingly. For calculaing venturi results use Eq. (52), correcting for velocity of approach by Eq. (62) and for gas compressibility by a factor from Table XXIII, page 272. Obtain coefficient of discharge from Fig. 160, page 280. Make complete calculation by Eq. (54), page 269, for one set of readings only, and compare with the results from Eq. (52).

e. Standard Pitot Tube.—Read discussion page 282. Use Eq. (52). There is no correction for velocity of approach or for gas compressibility, and the discharge coefficient is unity. But unless a traverse of the pipe is made, some assumption regarding the relation of average velocity to the velocity at the point of measurement must be made. Since the measurement was taken at the center of the pipe in this case, the corresponding velocity as calculated may be multiplied by 0.83 to obtain the approximate average velocity (see page 284).

f. Pitot Tube with Reversed Static Tube.—This form of pitot tube, in which the static tube is a duplicate of the impact tube but points directly opposite, or downstream, has an advantage for measuring low air velocities. For a given velocity the indicated differential pressure is greater than with a standard pitot tube on account of the suction effect upon the opening pointing downstream. The coefficient C is therefore not unity, and this coefficient is to be determined for the conditions of this experiment, using Eq. (52) as before. Use the factor 0.83 as in case *e*, and obtain Q from the venturi results.

g. Impact Tube.—Use Eq. (52) and a coefficient of discharge of 0.99 (see page 284).

4. Correcting Results to Standard Conditions.—There is no generally accepted standard of pressure, temperature, and moisture content at which to express the results of air and gas measurements. When dealing with air, the most logical procedure is to use a condition corresponding to a normal room condition. A convenient figure and the one perhaps most often used in this work is an air density of 0.075 lb per cu ft, corresponding to pressure of 14.7 lb per sq in., a temperature of 68°F, and a relative humidity of 36 per cent. The temperature of 68°F corresponds to the 20°C usually used by physicists. Other temperatures are sometimes used as standard (60, 65, or 70°), and other values of humidity are often specified, but the resulting standard density in any case is not far from 0.075 lb per cu ft.

In several of the formulas given above, it is necessary to use a value for the density of the air under test conditions, and, noting that the density is the reciprocal of the specific volume, this density may be computed from Boyle's and Charles's laws:

$$\rho = \rho_0 \frac{T_0 P}{T P_0} = 0.075 \frac{528}{T} \frac{P}{14.7}.$$

Very little error is introduced, however, in using the gas equation $PV = wRT$, with the value of $R = 53.34$ for dry air, as the humidity correction seldom reaches 1 per cent for ordinary atmospheric conditions and it is usually well under half that value.

5. Example of Calculations.—As an aid to overcoming the difficulties often encountered by the student in carrying through his initial calculation involving the flow formulas, the following example is offered:

An impact tube is located in the center of the jet of a flow nozzle discharging air from a low-pressure line to the atmosphere. The velocity pressure reading shown by the impact-tube gage is 0.65 in. of water, and the area of the nozzle throat is 4.33 sq in. The impact temperature is 80°F, and the barometer reads 29.10 in. Find the quantity of air discharged, expressing the result in terms of the volume at test conditions, the volume at standard conditions, and the weight of air per minute.

Solution.

$$Q = 60CA\sqrt{2gh}$$

$$A = \frac{4.33}{144}. \quad C = 0.99 \text{ (see page 279).} \quad g = 32.2$$

ρ = density of air at 80° and 29.10 in.

$$= 0.075 \times \frac{528}{540} \times \frac{29.1 \times 0.491}{14.7} = 0.0713 \text{ lb per cu ft}$$

or

$$\rho = \frac{PV}{RT}$$

$$= \frac{29.1 \times 0.491 \times 144 \times 1}{53.34 \times 540} = 0.0714 \text{ lb per cu ft.}$$

h = head in feet of air (at 80° and 29.10 in.) corresponding to 0.65 in. of water.

$$= \frac{0.65}{12} \times \frac{62.4}{0.0714} = 47.3 \text{ ft.}$$

Then

Q = volume of air at test conditions, cubic feet per minute

$$= 60 \times 0.99 \times \frac{4.33}{144}\sqrt{2 \times 32.2 \times 47.3}$$

$= 98.6$ cfm.

Q_0 = volume of air at standard conditions of 68°F and 14.7 lb abs, cubic feet per minute

$$= Q\frac{T_0}{T}\frac{P}{P_0}$$

$$= 98.6 \times \frac{528}{540} \times \frac{29.1 \times 0.491}{14.7}$$

$= 93.8$ cfm.

$$
\begin{aligned}
W &= \text{weight of air discharged per minute} \\
&= 98.6 \times 0.0714 \\
&= 93.8 \times 0.075 \\
&= 7.04 \text{ lb per min.}
\end{aligned}
$$

When several values are to be calculated in succession, the work may be much abbreviated by combining constants. For instance, in the above, assuming the barometric pressure to remain constant during several readings, and the temperature and velocity head to vary:

$$
\begin{aligned}
Q_0 &= 2{,}705\sqrt{\frac{h'}{T}} \\
&= 93.8 \text{ cfm,}
\end{aligned}
$$

where Q_0 = volume of air at standard conditions (see above).
h' = velocity head, inches of water.

V. NOTES AND PRECAUTIONS

1. Helical Flow.—A rotary or helical movement of the fluid on the upstream side of the flowmeter element produces an error through whirlpool or vortex action, and a greater flow than the actual will usually be indicated. The piping on the upstream side of a meter should always be critically inspected, as certain arrangements of elbows and tees will readily produce helical flow. In addition to this precaution, the A.S.M.E. and A.S.H.V.E. Test Codes recommend the use of transverse guide vanes ("egg-crate" type) in all tests:

"On pipe sizes of 2 in. and less, two crossing plates at right angles may be inserted in the pipe, provided they have a length in the axial direction of 1½ pipe diameters. For larger size pipes, it may be convenient to use a shorter length of guide vane, and if two vertical crossed by two horizontal plates be used, having an axial length equal to 1 pipe diameter, and spaced to give approximately even areas through all channels, rotary motion will be effectively prevented. These guide blades should be used in all cases; their presence will introduce no error and will ensure the absence of rotary motion. It has been found by experimental observation that rotary motion once started will continue with but slight reduction, even though a straightaway length of 20 or more diameters be employed."

2. Pulsating Flow.—The presence of pulsations in the fluid flow constitutes one of the most common sources of error in flowmeter measurement. In measurements of the discharge from reciprocating compressors or pumps, or of the supply of steam, gas, or air to reciprocating engines, this error may assume large proportions. Usually the meter will read high. Damping the pulsations of the meter gage will not eliminate the error. The following recommendations concerning the eliminations of this error are taken from a committee report:

"It is usually possible to eliminate pulsations of pressure by the use of storage reservoirs or receivers with the fluid flowing from one to another through a restriction. The flow originally pulsating is equalized and converted into a uniform flow at a lower pressure.

"This condition is fundamental, and there is at the present time no means of correcting the observations on pulsating flow to give accurate determination of quantity.

"With incompressible liquids, one or more air chambers may be employed with intermediate throttling between nozzle and source of pressure pulsation.

"The throttling and storage capacities must be such that there are no pulsations in the observations entering into the calculation of flow quantities.

"In general, two storage capacities and three points of throttling (*i.e.*, between the capacities, between the compressor and the first capacity, and between the second capacity and the measuring nozzle) will have greater effect in eliminating pulsations than the same total throttling in combination with a single storage-capacity equivalent to the two smaller capacities."

3. Selection of Metering Elements and Flow Equations.—It is important to distinguish between instruments for laboratory testing and those for continuous industrial service. This book deals particularly with the laboratory phase, but, as far as the venturi, nozzle, and orifice are concerned, these primary flow-measuring devices are the same in the industrial meters. (See "Fluid Meters, Part 3," A.S.M.E.) The secondary unit, or manometer, is very different for the two types of service, particularly since the industrial type must usually be recording and integrating as well as indicating. Moreover, the functional relation represented by the flow equation must be built into the secondary unit if it is to read directly in rate of flow. To accomplish these results the manometer liquid levels are usually transmitted to the reading element either mechanically by floats, cams, levers, or electrically by contact devices or transformer elements. These devices must of course be studied when an industrial meter is to be selected (see "Fluid Meters, Part 2," A.S.M.E.).

In selecting a head meter for laboratory testing, the necessary degree of accuracy must first be decided upon. The three elements in the selection should then be examined: (1) Primary element or differential producer: examined for cost, ease of installation, accuracy under the prevailing conditions, over-all head loss, capacity. (2) Secondary element or manometer: examined for range, sensitiveness, ease and speed of reading. (3) Flow equation: examined for applicability to the test conditions, ease of calculation, accuracy of equation and of discharge coefficients.

In general any differential producer may be used with any fluid, either for flow in a continuous pipe, for discharge from a pipe or from a reservoir to atmosphere, or for discharge from atmosphere into a reservoir. Pipe mounting is to be preferred, because most of the coefficients have been determined with this arrangement.

It is preferable to design the installation so that the simple hydraulic equation is applicable. With the proper coefficients, the hydraulic equation may usually be used with any compressible fluid and at any initial pressure, as long as the differential pressure is not over 10 per cent of the absolute initial pressure.

VI. FLUID FRICTION IN PIPES AND FITTINGS

EXPERIMENT 62. Pipe Friction Factors

I. PREFACE

The flow of fluids through pipes, fittings, valves, etc., results in a friction loss or drop in pressure. Similarly, the motion of a body within a fluid, such as the motion of a ship in the water, is resisted by a frictional "drag."

Both pipe friction and drag are produced partly by the viscous forces or skin friction and partly by the dissipation of energy in turbulence and eddies. When the viscous forces predominate, *i.e.*, below the critical Reynolds number, the pressure drop in a pipe is directly proportional to the velocity. When inertia forces predominate (Reynolds number above 3,000 approx), the friction loss is proportional to the initial kinetic energy of the stream, *i.e.*, to the average velocity head $V^2/2g$, or to the *square* of the velocity instead of to the first power. [See Eqs. (48) and (49) and discussion on page 260].

Friction factor may be obtained from Fig. 151, page 261. Figure 151 shows only four of many different degrees of pipe smoothness or "relative roughness" (ratio of height of surface projection diameter of pipe). The friction factor for other pipe sizes and materials may be estimated according to the degree of roughness.

The friction losses due to expanding or reducing fittings, as well as to elbows, tees, valves, and other devices, will depend on the exact nature of the flow path. In the average pipe-friction problem, these losses are most conveniently included by adding to the length of straight pipe an "equivalent length" which has the same resistance as the fitting. **Equivalent lengths of straight pipe,** for most common fittings and valves, are given in Fig. 152, page 262.

II. INSTRUCTIONS

1. *Setup.*—Long pipes of various kinds and sizes should be available, with approved static connections at each end and at each fitting (see page 273). For air flow it is preferable to use differential connection of manometers, but for liquids individual static pressure readings may be necessary. Valves, fittings, and transition pieces are to be included in the study.

2. *Procedure.*—Readings will be taken at five or more velocities for each pipe length, fitting, or test section. *Observations* will include pressure, temperature, and rate of flow by a metering device. Duplicate readings should be taken, at least 1 min apart. Pressures should be checked by having two static connections located 180 deg apart, at each plane or section. These should preferably be connected to separate gages.

3. *Calculations and Report.*—Results on pipe-friction tests should be plotted as shown in Fig. 151. An example of the Reynolds number calculation is given on page 260. The friction factor is calculated by Eq. (48) or (49). Discussion should include a comparison of experimental results with the data given in Figs. 151 and 152.

III. NOTES

If test pipes are too short, the entrance losses and the changes in velocity distribution near the entrance will affect the apparent friction factors.

Curves 2, 3, and 4 of Fig. 151 represent increasing degrees of roughness in the pipes, and curve 1 represents dead smooth surfaces such as those of drawn brass, tin, or lead pipes or glass tubing.

Instead of using equivalent lengths of straight pipe to account for losses in fittings and valves (Fig. 154), these losses are sometimes expressed in terms of the average velocity head at the entering section. This method neglects differences in surface finish within the fittings. Typical values of friction losses in terms of velocity head are

$$\text{Elbow} = \frac{0.5V^2}{2g},$$

and

$$\text{Tee} = \frac{1.0V^2}{2g}.$$

Most engineering applications involving air, gases, steam, or water are in the range of Reynolds numbers above the critical, *i.e.*, they involve turbulent flow. In many cases, such as in heat-transfer equipment, viscous or laminar flow is undesirable.

For water in a 1-in. standard pipe at room temperature, turbulent flow is usually obtained with velocities above 0.4 fps.

For air at room temperature and low pressure, turbulent flow is obtained with velocities above 5 fps in a 1-in. pipe or 0.5 fps in a 10-in. duct.

The pipe-friction equation and the friction-factor diagram are given various names by engineering writers, such as Fanning's formula, Darcy's equation, Stanton's diagram, etc. The Chezy formula is a slight modification of the same equation. Friction factors may also be approximated by the long-used formulas of Unwin, Babcock, and Fritzsche. Many correlations of formulas and friction factors have been made, but that of Pigott and Kemler is perhaps the most complete (*Mech. Eng.*, August, 1933, and *Trans. A.S.M.E.*, 1933).

VII. FRICTION DRAG OF IMMERSED BODIES

EXPERIMENT 63. Drag Coefficients

I. PREFACE

The resistance or drag of a body which is immersed in a fluid depends on several factors, including the shape of the body, the viscosity of the fluid, and the velocity of the body relative to the fluid. Completely submerged bodies such as airplanes, airships, and submarines present a different problem from partly submerged bodies such as boats and ships.

If the axis of the moving body is not parallel to the direction of motion, a cross-stream component is produced. This force is the **lift** of an airplane wing. It is also the force that propels a sailboat.

The two extreme cases of drag resistance are those of a flat plate parallel to the stream and a flat plate across the stream. These two cases also illustrate the two sources of drag resistance: (1) *skin friction*, and (2) *impact and eddies* or turbulence. Intermediate shapes such as cylinders, spheres, and streamlined bodies are retarded by both skin friction and eddy resistance. (See Fig. 202, page 493).

The *drag coefficient* is defined in terms of the kinetic energy or the velocity pressure of the approaching stream (compare with pipe-friction coefficient). It is a dimensionless coefficient equal to the ratio of the actual resistance pressure to the theoretical impact pressure. This definition of C_d as a ratio is indicated by Eq. (50), page 262. (Note that pressure = head × density.)

Values of the drag coefficient C_d for various shapes are given in the Appendix (Fig. 202). The drag coefficient is sometimes defined on the basis of total surface area as for plates or wings parallel to the stream. Mixed units may also be used. In comparing experimental data from various sources the exact definition of the drag coefficient should therefore always be ascertained.

II. INSTRUCTIONS

1. Setup.—The objects or models to be tested will be suspended in the testing section of a wind tunnel, with balances arranged to measure the total drag or force parallel to the air stream. Tests are to be run on a cylinder, a sphere, a flat plate, and on such other models as may be furnished by the instructor. The objects should be large enough so that the weighing error is negligible, but should not occupy more than 10 per cent of the wind-stream area. A good comparison is obtained if the projected area of two or more objects is the same. Thus the flat plate may be a disk of the same diameter as the sphere and the cylinder.

2. Procedure and Results.—Determine the drag resistance of each model, both perpendicular to the stream and parallel with it. Use at least five different velocities over the range available. Calculate the drag coefficients and plot curves similar to those of Fig. 202. In the case of the flat plate parallel with the stream, use the total surface area of both sides for *A* in Eq. (50). Make additional runs to define the curves at or near the critical velocities where the drag coefficient changes rapidly. Discuss results in comparison with those given in Fig. 202, or in other textbooks or references available. Compare the skin-friction drag coefficient of the flat plate parallel to the stream with the pipe-friction coefficient of Fig. 151.

CHAPTER VIII

PUMPS AND COMPRESSORS

PREFACE

The experiments of this chapter deal with pumps handling water and compressors handling air. The mechanical details and physical principles encountered will apply to many other kinds of machinery, however, among which might be mentioned:

Pumps for oils, tar, brine, sewage, chemical solutions, liquors, sirups, milk.
Compressors for ammonia, carbon dioxide, sulphur dioxide, natural gas.
Fans (low-pressure compressors) for fuel gases, for furnace gases, and for handling light materials in suspension in air.

In both construction and operating characteristics, pumps (for liquids) and compressors (for gases and vapors) are very much alike. In fact, certain types of compressors are often referred to as air "pumps" and vacuum "pumps."

Pumps and compressors appear in a great variety of designs, but most of them may be classified, on the basis of their principle of operation, into four classes:

1. Displacement pumps and compressors.
2. Centrifugal pumps and compressors (radial flow).
3. Screw or propeller type pumps and fans (axial flow).
4. Jet (or entrainment) pumps and compressors.

Displacement pumps and compressors deliver the fluid in successive isolated quantities, motion being imparted to the fluid either by a reciprocating piston or plunger, or by a rotating impeller such as a gear or a cam. Common examples of displacement pumps are the ordinary hand-operated well pump, the direct-acting steam-driven boiler-feed pump (Experiment 64) and the gear pumps used for circulating lubricating oil in an automobile engine. Most displacement compressors (air compressors, ammonia compressors, etc., Experiments 70, 71, and

82) are of the reciprocating type, although the use of the rotary compressor is increasing. The so-called "positive pressure blower" is often used when the discharge pressure is less than 20 lb gage.

Centrifugal pumps, fans, and compressors have displaced the other types for many kinds of service (Experiments 66 and 68). Outstanding among their advantages are: (1) a speed that permits of direct connection to electric motors and steam turbines; (2) compactness and low cost in large sizes; (3) uniformity of pressure and flow.

Propeller types of pumps and fans have the advantage of direct mounting on motor shaft and of very compact design for large volume and low pressure (Experiment 69).

In **jet pumps and compressors** a jet of fluid issuing from a nozzle entrains the fluid surrounding it. The velocity of the combined jet is then reduced by a diverging cone and the kinetic energy of velocity again converted into the potential energy of pressure. From the energy standpoint the efficiency of this process is low, but the device itself is simple and compact and is highly favored for certain uses (Experiment 65).

There are a few pumps and compressors not covered by the above classification. Among these might be mentioned the air lift, the pulsometer, the hydraulic ram, the Humphrey gas pump, the Taylor hydraulic compressor, the Jennings (Nash) liquid packing-ring type of compressor (Experiment 67). Each of these machines has a certain limited field of application.

EXPERIMENT 64. Test of Simple Direct-acting Steam Pump

I. OBJECT

As a performance test, the object of this experiment is to determine the duty, water rate, mechanical efficiency, and slippage of a simple direct-acting steam pump when operating at a constant speed and discharging against various heads. Since this is one of the first performance tests conducted by the student, certain other objects are to be kept in mind. Proficiency in the use of instruments and test equipment, teamwork on the test and in the calculating room, and the presentation of results in a creditable report—it is these accomplishments that measure the success of a test engineer.

II. PREFACE

Direct-acting steam pumps are classified as **simplex,** with one set of cylinders, and **duplex,** with two sets of cylinders set side by side. The duplex pump is the most common, the advantage of this type being the positive valve action, since the valve on one steam cylinder is actuated directly from the piston rod on the other, while the simplex pump must be equipped with some form of pilot valve and steam-actuated main valve. The duplex pump is sometimes provided with compound steam cylinders.

The water end of the direct-acting pump may use either a **piston** or a **plunger.** Soft **packing rings** or wide metal rings with circular water grooves are often used on the water pistons rather than the familiar narrow piston rings. Plunger pumps may be either **inside packed** or **outside packed,** the latter being favored on account of the ease of packing.

Pump **valves** are usually flat disks of soft or hard rubber or composition, though leather-faced valves and metallic valves are sometimes used. An **air chamber** is provided on the pump discharge in order to reduce vibration and noise at higher speeds and to reduce the pulsations in the flow of water.

The **piston speed** of direct-acting pumps should ordinarily not exceed 100 ft per min for general service, and boiler-feed pumps are usually operated below 50 ft per min. Long-stroke pumps may of course be operated at higher piston speeds than short-stroke pumps.

The direct-acting pump is sturdy and reliable, but it is wasteful of steam and is therefore seldom used in large sizes at the present time, except as a stand-by unit.

III. PREPARATIONS

1. Apparatus.—The setup of this experiment consists of a simplex pump mounted above a suction tank or sump and discharging through a throttle valve to weighing tanks mounted on platform scales. The steam end of the pump is provided with indicator connections and the exhaust is piped to a small surface condenser. The following organization of the test crew is suggested:

Position	Duties	Equipment
1	Foreman and timekeeper. Adjust throttle valves, read gages and stroke counter	Two 100-lb. double-spring gages. Stroke counter
2	Weigh condensate. Make data sheet and collect all data	Two 12-qt. pails. Small platform scale and weights. Data board
3	Weigh water pumped. Get pump dimensions	Weights for platform scales. Steel scale and calipers
4	Take indicator diagrams, water temperature, and steam quality	Indicator and cards. 120-deg. (or 220-deg.) thermometer. Calorimeter
5	Planimeter and calculations.........	Planimetor and slide rule
6	Planimeter and calculations.........	Planimeter and slide rule

IV. INSTRUCTIONS

1. Procedure.—The entire test is to be run at a constant pump speed. A moderate speed should be selected (see Preface), and one that can be maintained when pumping against the maximum pressure. It is suggested that the maximum discharge pressure shall correspond to the average available steam pressure.

The foreman will set the steam- and water-throttle valves before each run, to obtain the desired speed and discharge pressure.

2. Determinations.—Make six runs of 20 min each, taking all readings every 5 min. Use approximately equal increments of pressure.

3. Results.—Data and results of the test are to be reported on Form 64. Plot curves of duty, water rate based on indicated horsepower, water rate based on brake horsepower, and slippage (all on the ordinates), against discharge pressure. Discuss accuracy of results and compare the economy of this unit with the economy of pumps and pumping engines of other sizes and types. The A.S.M.E. Test Code on "Reciprocating Steam-driven Displacement Pumps" calls for the calculation of thermal efficiency and of engine efficiency or Rankine-cycle efficiency ratio in addition to the performance results listed on Form 64.

Form 64.—Results of Steam-pump Test

Maker's name.........................Nominal size..............
Dimensions:
Diameter of steam cylinder.....in. Diameter of rods.....in.
Diameter of water cylinder.....in. Diameter of steam pipe....in.
Diameter of suction pipe.......in. Diameter of exhaust pipe....in.
Diameter of discharge pipe.....in. Volume of air chamber.... cu. in.

Item number	Item	Units	Run No.				
			1	2	3	4	etc.
1	Duration of run	Min.					
2	Total number of strokes						
3	Single strokes per min						
4	Average length of stroke	Ft.					
5	Piston speed	Ft. per min.					
6	Steam pressure, gage	Lb.					
7	Quality of steam	Per cent					
8	Total condensate	Lb.					
9	Total dry steam	Lb.					
10	Discharge pressure, gage	Lb.					
11	Discharge head	Ft.					
12	Suction head	Ft.					
13	Total head	Ft.					
14	Temperature of water	Deg. fahr.					
15	Water pumped per min	Lb.					
16	Water pumped per min	Cu. ft.					
17	Plunger displacement per min	Cu. ft.					
18	Slippage	Per cent					
19	Indicated horsepower steam cylinder, head end	Hp.					
20	Indicated horsepower steam cylinder, water end	Hp.					
21	Indicated horsepower steam cylinder, total	Hp.					
22	Water horsepower (useful work)	Hp.					
23	Mechanical efficiency	Per cent					
24	Dry steam per i. hp-hr	Lb.					
25	Dry steam per water-hp-hr	Lb.					
26	Duty per 1,000,000 B.t.u	Ft -lb.					
27	Duty per 1,000 lb. steam	Ft -lb.					
28	Capacity per 24 hr	Gal.					

Date:.............. Observers:..............

These are defined in Experiment 79, page 400, and may be added if desired.

EXPERIMENT 65. Test of Steam Injector

I. PREFACE

The injector is a common example of the jet type of pump. This device is probably best known as a locomotive boiler feeder. It is also used to some extent in small stationary plants, especially as a reserve or spare unit. For best service the injector should be designed for the conditions of operation under which it is to be used. There are several types of injectors, as single tube and double tube, lifting and nonlifting, manual and automatic starting, etc. As a pump the injector is very wasteful of steam, but it is an excellent combined pump and feed-water heater and it is used where such a combination is desired and exhaust steam is not available for feed-water heating. The injector will not pump hot water, unless the water is supplied under pressure, and it does not work well under extreme variations of load.

The procedure for an injector test and the results obtained are much the same as those for any other pump test. There are no test-code rules covering this procedure, as commercial tests of these small units are seldom made. The injector test affords a good opportunity, however, for a study of energy distribution or heat balance in a typical heat-energy conversion process and at the same time illustrates the principles involved in the operation of a wide variety of jet appliances. Moreover, on account of the fact that the steam consumption of the injector cannot be weighed directly, *the test is one in which very careful manipulation is necessary to secure consistent results.* The weight of steam supplied is obtained by two methods, by calculation from the heat balance and by the difference between the weight of water supplied *to* the injector and the weight of water delivered *by* it.

II. INSTRUCTIONS

1. Setup of Apparatus.—The injector is mounted over a suction tank, and this in turn is mounted on platform scales. The discharge is piped to another weighing tank. If small tanks are used, two tanks should be provided for each service and the proper valves placed for changing from one tank to the other. A steam calorimeter and a gage are to be provided on the steam line, a gage on the discharge line, and thermometers in the suction and the discharge.

2. Starting the Injector.—The method of starting depends to some extent upon the type of injector to be tested. Fill the suction tank with cold water and see that the suction line is open. With the injector valve closed, open the main steam valve and the injector discharge valve. Open the injector valve, slowly at first. Throttle the discharge valve until the desired discharge pressure is obtained. If the injector "breaks" and loses its water, the injector steam valve should be closed at once, for, if steam is allowed to blow through until the injector becomes hot, it will fail to pick up water until it has been cooled again.

3. Determinations.—Make at least four sets of duplicate runs at various discharge pressures, keeping all other conditions constant. If any of the duplicate runs do not check closely, they should be repeated. Before dismantling the apparatus, calculate pounds of water discharged per pound of steam supplied, and plot this quantity (on the ordinates) against discharge pressure. If erratic points are found, the tests upon which they are based should be repeated. Make additional tests to determine the limits of operation of the injector, as follows: (*a*) Ratio of maximum discharge pressure to steam pressure; (*b*) minimum steam pressure to maintain a given discharge pressure; (*c*) minimum steam pressure to establish suction.

4. Results and Calculations.—Results of the tests will be similar to those calculated for Experiment 64, page 308, with the addition of a "heat balance." The total head includes the velocity head. The energy for pumping water may be taken as the energy required for pumping the total water supplied, against the total dynamic head. The total energy supplied by the steam is taken as the total energy for pumping and heating the supply water, heat-radiation losses being neglected.

The steam consumption is to be arrived at by difference in weights of the water supplied and the water discharged and also on the basis of the weight of steam shown by computation of the heat balance, radiation being neglected. If the heat supplied to the injector (above 32°) is equated to the heat delivered in the discharge stream plus work done plus radiation losses, there is but one unknown, *i.e.*, the weight of steam supplied.

Curves should be plotted, using discharge pressures as abscissas, showing the variation in the following: (1) water delivered per pound of steam supplied; (2) thermal efficiency of the injector as a pump; (3) steam consumption in pounds of dry steam per water-horsepower hour (both methods).

EXPERIMENT 66. Test of Centrifugal Pump

I. PREFACE

An ample supply of water is evidently one of the first essentials to progress in civilization, and the development of water-storage and distribution systems has therefore always ranked as a major engineering problem. Many crude mechanical devices for "pumping" water have been in use for thousands of years, but the evolution of these machines has now reached a point where a single pump may handle millions of gallons of water daily and the over-all loss in pumping may amount to less than 20 per cent of the energy supplied to the machine.

In some respects the most highly developed form of pumping machine is the centrifugal pump. Practically all pumps now being installed for city water service or for industrial or institutional water supply are centrifugal pumps. In irrigation and drainage work, for suction dredging, for handling sewage, for fire service, and in the central station or large power plant for boiler feed, condenser circulating water and house service, the centrifugal pump is preferred over the other types.

The centrifugal pump is a very simple machine in that it consists of only two parts, the **casing** and the **impeller.** The impeller rotates within the casing, and mounted on the latter are the bearings for the impeller shaft and the stuffing boxes or glands for preventing leakage along the shaft.

Centrifugal pumps may be classified as *horizontal or vertical* (indicating the position of the shaft), *volute or turbine, single stage or multistage, single suction or double suction, open or closed impeller.*

A **single-stage pump** is one that uses a single impeller for raising the pressure of the liquid from suction pressure to discharge pressure. For heads above 100 to 200 ft (depending on the speed), it is the usual practice to use two or more impellers[1] connected in *series* (Fig. 171). In high-

[1] With a 3,600-rpm drive, a single-stage pump may be designed for a discharge head as high as 650 ft.

pressure boiler-feed pumps as many as five or six stages are used. For the purpose of increasing capacity, pump stages (or independent pumps) are sometimes connected in *parallel.* Occasionally, dual-purpose pump sets are constructed for operation either in series or in parallel, as the service demands. An example of such a set is a combination service and fire pump composed of two pumps operated singly or in parallel for ordinary service but connected in series for fire service.

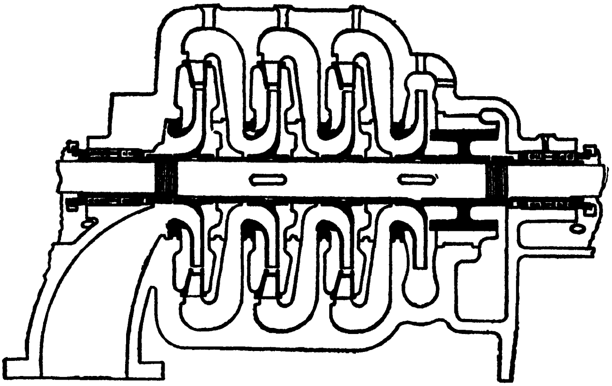

FIG. 171.—Multistage centrifugal pump.

Most single-stage centrifugal pump casings are built in the form of a spiral or volute, whereas multistage pumps often have cylindrical casings. **Volute pumps** by their geometrical form provide a gradually increasing area of the liquid passage approaching the discharge, but in order to secure this effect in a pump with a cylindrical casing, **diffusion vanes** are inserted in the casing. This type of pump is called a "turbine pump." (A new design has recently been offered which has neither volute nor diffusion vanes.)

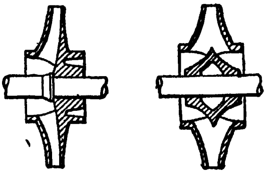

FIG. 172.—Single- and double-suction impellers.

Pump impellers are built for either **single suction** or **double suction,** *i.e.*, they may take water from one side only or from both sides or faces of the wheel (see Fig. 172). The double-suction pump has the advantage of hydraulic balance and consequent freedom from axial thrust. Impellers are further classified as *open* or *closed.* Open impellers are simpler but are less efficient. They are used on the cheaper single-stage pumps and also for such special services as handling paper pulp or pumping sewage. In a closed impeller the vanes or blades are enclosed on both sides by disks or shroud rings. When closed impellers are used, it is necessary to provide small running clearances outside the impeller to prevent backflow of the water around the impeller from discharge to suction. For this purpose **wearing rings** or sealing rings are used. On high-grade pumps these are renewable.

Stuffing boxes are provided to seal the rotating shaft where it passes through the casing. The stuffing boxes subjected to the suction pressure of the pump are usually water sealed, the water being supplied under pressure from the discharge casing. This is desirable because any inleakage of air along the pump shaft or shaft sleeve is likely to cause the pump to "lose its suction" and fail to operate. The **shaft sleeve** is a protecting covering of noncorrosive metal, fitted over the steel shaft to protect it from wear, rust, and corrosion. Chromium plating is also used by some manufacturers to protect the shaft. Pump **bearings** are usually either ring-oiled sleeve bearings or ball bearings with grease lubrication.

Principle of Operation.—If the casing and impeller of a centrifugal pump, fan, or compressor stand full of the fluid that is to be handled, and the impeller is set in motion, the centrifugal force of the rotation will tend to throw outward into the casing the fluid that is contained in the impeller. If an intake and a discharge are provided, the action will be continuous and a stream of the fluid will be put into motion. If gradually expanding passages are provided in the casing, the initial velocity of the fluid leaving the impeller may be reduced, with a resultant increase in pressure. In other words, the velocity head may be converted into pressure head, or the kinetic energy into potential energy. This head conversion takes place in accordance with the familiar law $V = \sqrt{2gh}$. The theoretical maximum head is of course never realized in the actual machine because of various **hydraulic losses.**[1] These losses together with the **mechanical losses** also determine the efficiency of the pump. Among the hydraulic losses may be mentioned: (1) inlet or suction loss by eddies caused from sudden changes in the size and direction of the flow passages; (2) impeller friction and shock losses; (3) impeller exit losses; (4) casing losses due to eddy currents and friction; (5) backflow or leakage from high-head to low-head regions by short-circuit paths. The mechanical losses include bearing losses and stuffing-box losses.

The magnitude of the total losses and the consequent efficiency of the pump depend in part upon the perfection of the design and in part upon the operating conditions. Some of the cheaper designs of pumps are not capable of maximum efficiencies above 55 or 60 per cent, whereas the better designs may show efficiencies as high as 85 per cent. But it is also to be noted that a pump that may be capable of operating at a very high efficiency under one set of conditions may show a very low efficiency under another set of conditions (see Fig. 173). Although a centrifugal pump, therefore, will operate over a wide range of capacities, heads, and speeds, it must be designed for its usual operating condition if a high degree of efficiency is to be maintained.

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[1] There is some disagreement as to whether this theoretical maximum head should be taken as $V^2/2g$ or V^2/g, for an impeller with radial vanes.

Laws of Pump Performance.—The following laws approximately define the performance of a centrifugal pump (or fan or compressor) under various conditions (see also Table XXVI, page 324):

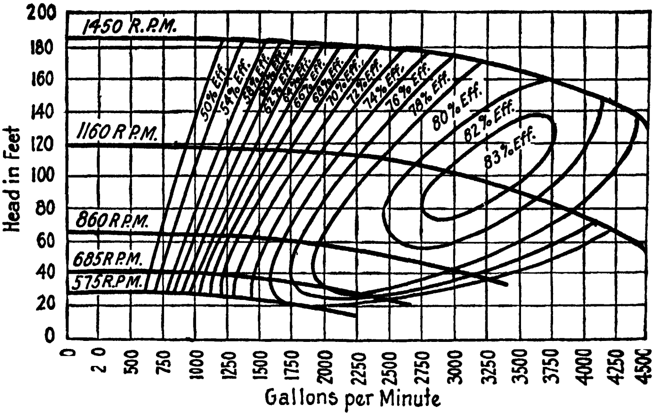


FIG. 173.—Variable-speed performance of a centrifugal pump, showing iso-efficiency curves. (One impeller diameter.)

1. With a given pump and piping system, **the capacity of the pump varies directly as the speed, the total head varies as the square of the speed, and the horsepower to drive varies as the cube of the speed.** These laws are applicable when the external resistances are flow resistances only. If a static head exists (*i.e.*, if the pump discharge is under a head of water when the impeller is at rest), they must be modified somewhat. From the equation $h = V^2/2g$ it would be expected that the head would vary as the square of the speed. Since the work done is directly proportional to the product of head and capacity (horsepower = $PV/33{,}000$), it follows that the power required would vary as the cube of the speed. For comparatively small variations in speed (say less than 20 per cent above or below normal), the

efficiency curve of a centrifugal pump is not greatly affected (see Fig. 173).

2. For a constant speed in revolutions per minute the capacity varies approximately in direct proportion to the diameter of the impeller, the head varies as the square of the impeller diameter, and the power required varies as the cube of the impeller diameter. These relations follow from the previous laws and from the fact that the peripheral speed varies as the diameter of the impeller.

Although a centrifugal pump may be purchased to meet almost any specified head and capacity condition, the manufacturers carry only a few sizes of pump casings. Each casing serves for several conditions by the use of impellers of various sizes.

In the present experiment a complete set of performance curves such as those shown in Fig. 173 will not be obtained. A sufficient number of runs will be made, however, to determine the condition of maximum efficiency for the pump under test, and to ascertain the effect of varying the external resistance to flow, varying the speed, and changing the size of impeller used in a given pump casing. In order to demonstrate the variation of pump performance with speed and with impeller size, it is necessary to furnish some fixed and easily duplicated condition of external resistance to flow, in addition to the "full-open" and "closed" positions of the discharge valve. It is difficult to duplicate exactly the discharge-valve setting for an intermediate condition. A satisfactory method for introducing a fixed resistance is to install a by-pass around the main discharge valve, equipping the by-pass with a fixed orifice and with a shutoff valve. In the following discussion it will be assumed that the test apparatus is equipped with such a by-pass and that the orifice used in the same is of such a size that it will load the pump to approximately rated capacity at rated head when the pump is running at rated speed.

II. INSTRUCTIONS

1. Determinations.—With each of the impellers available, make at least three sets of runs at different speeds. At each speed five or more determinations of head, capacity, and horsepower should be made. Use approximately equal increments of capacity, varying the same by throttling the discharge valve.

Full-open valve, fully closed, and three or more intermediate valve settings should be used, and in addition a run should be made with the fixed by-pass open and the main valve closed.

Form 66.—Results of Centrifugal-pump Test

Maker's name........................Maker's type and size..........
Size of suction pipe....Size of discharge pipe....Diameter of impeller..
Actual area of suction pipe........Actual area of discharge pipe.......
Rated capacity............Rated head...........Rated speed........
Type: horizontal or vertical...Volute or turbine....Number of stages..
Open or closed impeller..............Single or double inlet...........

Item number	Item	Units	Run No.			
			1	2	3	4
1	Duration of run...............	Min.				
2	Speed.........................	R.p.m.				
3	Discharge pressure............	Lb.				
4	Discharge head...............	Ft.				
5	Suction, by gage..............	In. Hg				
6	Suction head..................	Ft.				
7	Vertical distance between gages.	Ft.				
8	Velocity head in suction pipe...	Ft.				
9	Velocity head in discharge pipe.	Ft.				
10	Total dynamic head...........	Ft.				
11	Temperature of water..........	Deg. fahr.				
12	Pump capacity, by weight......	G.p.m.				
13	Pump capacity, by volume.....	G.p.m.				
14	Pump capacity, by venturi.....	G.p.m.				
15	Pump capacity, by weir........	G.p.m.				
16	Pump capacity................	G.p.m.				
17	Water horsepower.............	Hp.				
18	Power to drive................	Hp.				
19	Pump efficiency...............	Per cent				

Date................ Observers:.

2. Results.—Data and results should be tabulated on Form 66. The "total dynamic head" is the total head produced by the pump and is equal to the difference between the total (static plus velocity) heads at the discharge and suction nozzles, taking into account any difference between the elevations of the gages used. Measurement of water should be by weight or volume, if possible, and these results should then be used in the calculations. If rate-of-flow measurements must be resorted to, and both the venturi meter and weir are available, the averages of these results should be used in the calculations. (For methods of calculating venturi and weir results see Chap. VII.)

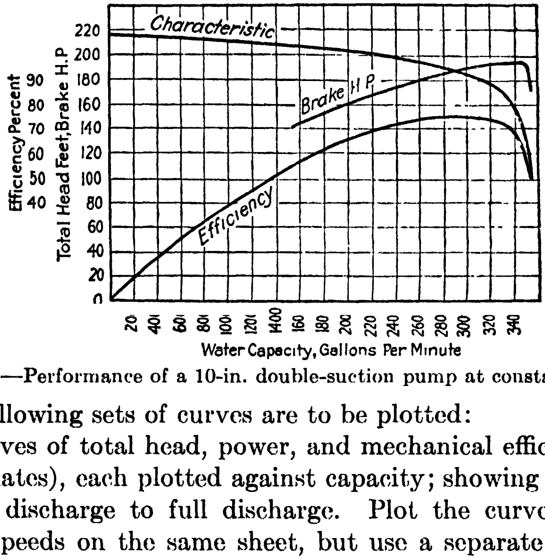


FIG. 174.—Performance of a 10-in. double-suction pump at constant speed.

The following sets of curves are to be plotted:

1. Curves of total head, power, and mechanical efficiency (on the ordinates), each plotted against capacity; showing all values from no discharge to full discharge. Plot the curves for the various speeds on the same sheet, but use a separate sheet for each impeller size (see Fig. 174, also Experiment 68).

2. Curves of total head, power, capacity, and mechanical efficiency (on the ordinates), each plotted against speed; showing values for the fixed orifice or by-pass only. Use a separate sheet for each impeller size.

3. Curves of total head, power, capacity, and mechanical efficiency (on the ordinates), each plotted against diameter of impeller; showing only those values for the fixed orifice or by-pass and for normal or rated speed.

Give complete discussion of test results as shown by the curves.

III. OPERATING PRECAUTIONS

Starting and Stopping.—Before starting, fill suction pipe and pump casing with water and close discharge valve. After starting motor, open discharge valve gradually to obtain desired capacity. Before stopping motor, close discharge valve.

From the discussion under Principle of Operation, above, it is evident that a centrifugal pump will refuse to operate unless the impeller is filled with water before starting. There are various ways of **priming** a pump. If an external supply of water is provided, a "foot valve" is placed at the bottom of the suction pipe to hold the priming water in the pump. Another device is a "priming ejector" or other vacuum pump which provides a vacuum on the pump for starting and thus fills the suction pipe and the pump. The air-vent cocks in the top of the casing should be opened when priming from a pressure source, and also after starting, to ensure the discharging of all air from the casing.

Do not draw the stuffing-box glands tight. Water dripping slowly from the stuffing boxes when the pump is operating keeps the packing in good condition.

The pump discharge valve may be fully closed while the pump is in operation, but it should not be left closed long enough to allow the water in the casing to get hot.

Care must be taken to stop any leaks in the gage connecting piping. This piping should always be full of water and free from air pockets.

The location and method of attaching gages are important; this subject is well covered by a series of 12 figures in the A.S.M.E. Code.

EXPERIMENT 67. Test of Return-line Vacuum Heating Pump

I. PREFACE

The return-line vacuum pump is a combined water and air pump for removing condensate and air from the return or drain lines of large vacuum heating systems. The unit usually consists of two pumps which function independently, and the discussion and instructions given in Experiment 66, page 314, apply to the centrifugal water pump of this unit.

According to the A.S.H.V.E. Test Code, the **standard basis of rating** a vacuum heating pump calls for the simultaneous air and water capacity of the unit (in cfm and gpm, respectively), and the power input, when operating with a vacuum of 5.5 in. Hg at the pump inlet, handling air and water at a temperature of 160°F, and discharging the water against a pressure of 20 lb gage.

II. INSTRUCTIONS

The essential measurements are speed, torque, head and capacity of the water side, and head and capacity of the air side.

The air handled is to be measured by an orifice or nozzle, and the air quantity may be calculated by Eq. (52) or (60), page 269.

Using the conditions for the standard basis of rating, make at least three runs of 20 min duration or more each, with readings every 5 min. Power and capacity results for these runs should check within 2 per cent. Another set of runs should be made with constant speed, constant vacuum, and variable discharge secured by throttling the discharge valve. At least five points should be taken in this test including wide-open valve and fully closed valve.

The report should include a comparison with manufacturer's rated performance, analysis of accuracy of tests, and a discussion of the performance of the unit under various operating conditions.

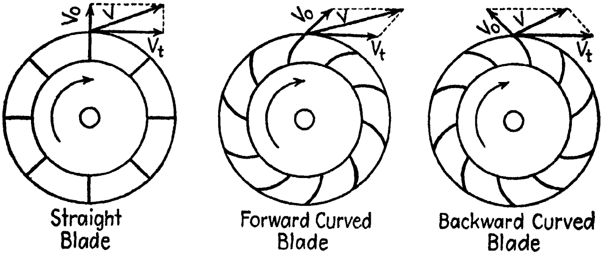
EXPERIMENT 68. Test of Centrifugal Fan

I. PREFACE

Of the various types of "air compressors," the centrifugal fan is the machine most commonly employed for handling air and gases at pressures less than 1 lb per sq in. The centrifugal fan has therefore a wide variety of applications, most of which might be included under the following heads: (1) ventilation; (2) heating and cooling; (3) drying, evaporating, and humidifying; (4) conveying; (5) draft or blast for boiler, metallurgical, or other fires.

Commercial performance tests of fans are most often concerned with the volume of air delivered per minute under specified conditions, *i.e.*, with the "fan capacity." Static pressure, as one of the specified conditions, is also measured. Horsepower input is measured if practicable, but in tests of installed fans it frequently can be only approximated. A complete fan test cannot be made, however, without accurate power measurements. The performance figures arrived at by test are compared with those given in the fan manufacturers' "performance tables," see page 330.

A "Standard Test Code" for centrifugal and propeller fans has been formulated jointly by the American Society of Heating and Ventilating Engineers and the National Association of Fan Manufacturers. This Code specifies standard terms, instruments, methods, and form of report. Commercial rating tests and acceptance tests are made in accordance with the latest edition of the A.S.H.V.E. Standard Test Code. The A.S.M.E. Committee on Power Test Codes is also cooperating to improve and extend the methods of fan and blower testing.



Straight Blade Forward Curved Blade Backward Curved Blade

FIG. 175.—Vector diagrams for three types of fan blades.

Laws of Fan Performance.—The centrifugal fan, like the centrifugal pump, is a "steady-flow machine," and as such it has more in common with a steam turbine than it has in common with a piston air compressor or a positive-pressure blower. Thus a vector diagram may be drawn to represent the velocities of blades and air, and such a diagram (Fig. 175) shows that a fan with forward-curved blades may be expected to produce a higher pressure than one with radial blades, if both operate at the same tip speed. But the actual pressures produced with a given fan depend also upon the size of outlet available, or in other words, upon the resistance of the connected ductwork. Figure 178 gives a typical set of test curves showing the performance for a given fan over the entire range of duct resistances, from wide open to blocked tight. Similar curves for other types of fan wheels are briefly illustrated in Fig. 176.

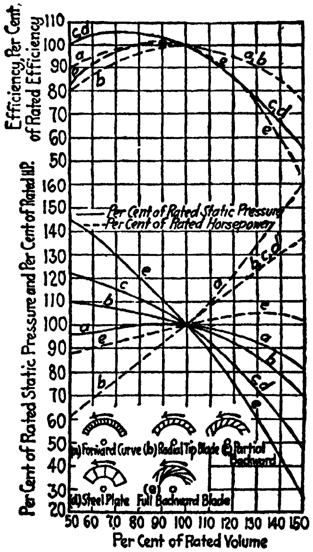


FIG. 176.—Characteristics of various types of fan wheels. (*From Marks, "Mechanical Engineers' Handbook."*)

Fan performance is dependent upon speed of rotation and upon air density as well as upon fan design and duct resistance. **Table XXVI is a condensed statement of the laws of performance of centrifugal fans, blowers, and pumps.** These laws are used particularly for determining: (*a*) Performance of a fan at one speed, calculated from tests at another speed. (*b*) Performance to be expected from a large fan, calculated from tests on a smaller but geometrically similar fan. (*c*) Performance changes to be expected with changes in air temperature or changes in barometer reading.

TABLE XXVI.—PERFORMANCE LAWS OF CENTRIFUGAL FANS AND BLOWERS

Applicable also to centrifugal pumps and to propeller fans and pumps

A. For a given fan, operating on the same unchanged piping system with air at the same density:

1. Volume varies directly as rpm.
2. Pressure varies as [rpm]^2.
3. Power to drive varies as [rpm]^3.
4. Pressure varies as [volume]^2.
5. Power varies as [volume]^3.

B. For a series of geometrically similar fans operating at the same tip speed, against the same (proportional) piping system and with air at the same density:

6. Volume varies as [wheel diameter]^2.
7. Pressure remains constant.
8. Power varies as [wheel diameter]^2.

C. For a given fan, operating at constant speed on the same unchanged piping system:

9. Volume remains constant with changes in air density.
10. Pressure varies directly as the air density.
11. Power varies directly as the air density.

Some care should be exercised in using the fan laws given in Table XXVI, particularly when applying them to obtain values for power and efficiency of small fans. Departures from exact geometrical similarity are often made for manufacturing reasons, and all dimensions should be checked before similarity is assumed. For performance at different speeds, exact values of the exponents may be readily obtained by plotting test values of pressure, volume and horsepower, against speed, on logarithmic coordinates (for tests at constant orifice). The exponent for the pressure curve is frequently greater than 2, and the exponent for the horsepower curve may be considerably less than 3, especially if bearing and belt losses are included in the power to drive.

II. PREPARATIONS

1. Apparatus.—The fan should be equipped with a round test pipe of minimum dimensions shown in Fig. 177, with straightening vanes[1] and throttling orifices.

A pitot tube is specified for air measurement (see page 282), and it should preferably be connected so that velocity pressure, static pressure, and total pressure can each be read for checking (Fig. 163; see also pages 13, and 283). The pitot tube station must be preceded by at least 7.5 diameters of straight pipe, and a traverse of 20 readings is specified by the Code. The pitot-tube stations are in equal concentric areas and are laid out on two diameters, as shown in Fig. 162 and Table XXV, page 282.

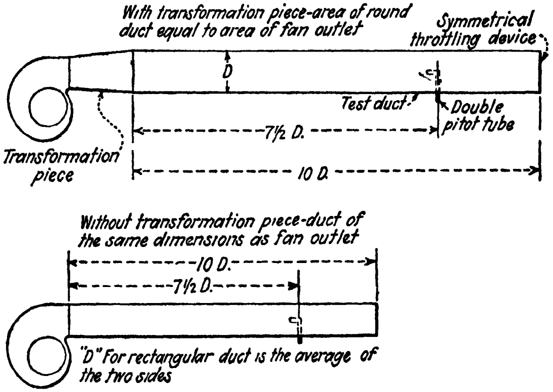

With transformation piece-area of round duct equal to area of fan outlet
Symmetrical throttling device
D
Test duct
Double pitot tube
7½ D.
10 D.
Transformation piece
Without transformation piece-duct of the same dimensions as fan outlet
10 D.
7½ D.
"D" for rectangular duct is the average of the two sides

FIG. 177.—Dimensions of test pipe and location of pitot tube. (*A.S.H.V.E. Standard Test Code.*)

For power measurements a cradle dynamometer is preferred (Fig. 82), but a calibrated electric motor may be used (see typical calibration curves, page 103, Fig. 81 and Table IX.)

Portable instruments include an accurate speed counter, thermometers for air temperature, and scale weights or electrical instruments for the dynamometer or motor.

2. Setup.—(1) Attach manometers to a solid mounting, level accurately and fill with indicating liquid to zero marks. (2) Connect pitot tube so that independent readings of static, total, and velocity pressures may be made. (3) Rule up a "spacer board" for pitot-tube traverse according to the dimensions in Fig. 162, page 282, or if test pipe is too small to allow 20 readings, use Table XXV. The spacer board is to be placed against the duct next to the pitot tube, and the traverse is made by lining up each one of the marks on the spacer board with a reference mark on the pitot tube. (4) Adjust dynamometer, and place such instruments as are necessary for its operation (electrical instruments, scales, etc.). (5) Inspect and oil all bearings. (6) Prepare data sheets.

---

[1] The straightener shall consist of square egg-crate type cells, each 0.075 to 0.15 pipe diameter on a side. Length of straightener to be three times the side of each square, and location to be 6 pipe diameters from fan end of straight test pipe.

III. INSTRUCTIONS

1. Test Procedure.—*Test* 1: Take dimensions of fan, test pipe, and orifices. Observe and record zero reading of each manometer. With full outlet of discharge pipe, take the following readings: (1) speed of fan (kept as nearly constant as possible); (2) air temperatures at inlet and discharge; (3) dynamometer readings; (4) total, static, and velocity pressures (pitot-tube traverse, 20 readings of each pressure). Repeat the above readings for each size of orifice and for no discharge, using the same fan speed throughout. Check zero readings of gages after completing the test with each orifice. Read barometer before and after taking pitot-tube readings.

Test 2: Using an orifice representing about three-fourths of full outlet area (or 75 to 90 per cent of pipe diameter), repeat the test runs at four other fan speeds in addition to the one used in Test 1. This will give 5 values of pressure, volume, and horsepower, each to be plotted against speed, the duct resistance remaining constant. The test may be abbreviated if necessary, by using the orifice as a metering device and omitting the pitot-tube readings. In that case the readings should be made in triplicate for checking.

Other methods of air measurement may be used as a supplement to the pitot tube, depending on the laboratory setup. The fan test furnishes a good opportunity for obtaining test experience with such metering devices as the flow nozzle, the venturi tube, or the anemometer.

2. Results and Calculations.—It is customary to report fan test results on the basis of "standard air" (0.075 lb per cu ft).

It is also necessary to correct for the duct friction or pressure loss between fan outlet and pitot tube, and for the slight accidental variations in fan speed. Items 1 to 10 in Form 68 represent the actual test conditions. Since the duct friction represents a drop in static pressure, the friction correction is added to the static and the total pressures (items 11 to 13). Correction factors for fan speed and for air density are next obtained and applied to items 12, 13, 10, and 9 to obtain the corrected values of items 14, 15, 16, and 17, respectively.

The friction correction, item 11, is obtained by using the pipe-friction equation specified by the Code (see page 261):

$$\Delta P = 0.02 \frac{L + 4}{D} (VP),$$

where L is the distance from fan outlet to pitot-tube location in feet, D is the duct diameter in feet and VP is the average velocity pressure as observed. Four duct diameters are added to account for the friction in the straightening vanes. The correction ΔP is added to the observed average pressures, to obtain items 12 and 13.

Correction factors for fan speed are based on the laws in Table XXVI which state that volume varies as the first power, pressure as the square and horsepower to drive as the cube of the fan rpm. **Corrections for density** are similarly based on the laws which state that pressure and horsepower vary directly with the air density, the speed and capacity being constant.

Fan capacity is calculated by Eq. 52, page 270. For ordinary tests it is sufficiently accurate to use the average of the observed velocity pressures as the Δh under the radical in this equation. The more accurate method, which is required by the Code, is to compute the velocity corresponding to each measured velocity pressure (20 values for each traverse, by the equation $V = \sqrt{2g\Delta h}$), then to average the resulting velocities and obtain the volume by the equation $Q = AV$.

Air horsepower is based on the fundamental fluid horsepower equation neglecting compressibility: $PV/33{,}000 = \text{hp}$, where P is total pressure, pounds per square foot and V is cubic feet per minute. Mechanical efficiency is the ratio of air horsepower to horsepower input. Static efficiency is mechanical efficiency multiplied by the ratio of static pressure to total pressure.

Form 68.—Results of Centrifugal-fan Test

Type of fan........................ .....Wheel diameter....................in.
Manufactured by.......................Located at...........................
Test pipe: Length............ft Diam ....in. Area = *A*............sq ft

Item number	Item	Units	Data and results	
1	Orifice area	Sq in.		
2	Percentage of full outlet area	Per cent		
3	Velocity pressure	In water		
4	Static pressure	In. water		
5	Total pressure	In. water		
6	Fan speed	Rpm		
7	Air density in discharge duct	Lb per cu ft		
8	Air density at fan inlet	Lb per cu ft		
9	Horsepower input to fan	Hp		
10	Fan capacity (volume)	Cfm		
11	Friction correction	In. water		
12	Static pressure at fan outlet	In. water		
13	Total pressure at fan outlet	In. water		
14	Static pressure, corrected	In. water		
15	Total pressure, corrected	In. water		
16	Fan capacity, corrected	Cfm		
17	Horsepower input, corrected	Hp		
18	Air horsepower	Hp		
19	Mechanical efficiency	Per cent		
20	Static efficiency	Per cent		

Date..................... Observers:

3. Report and Discussion: The curve sheet showing the results of Test 1 will be similar to Fig. 178. Results of Test 2 are to be plotted on logarithmic coordinates, with rpm as abscissas. Three straight lines will be obtained, the volume line having a slope of approximately 1.0, the pressure line a slope of 2.0 or slightly more, and the horsepower line a slope of 3.0 or slightly less.

Report will be submitted in the usual form, with a discussion of test results and conclusions (page 4). If possible, a comparison should be made with the manufacturer's tables and curves for the particular fan tested, (ask instructor for bulletins). Data in the following section, and Table XXVI, will furnish additional material for comparisons and discussion.

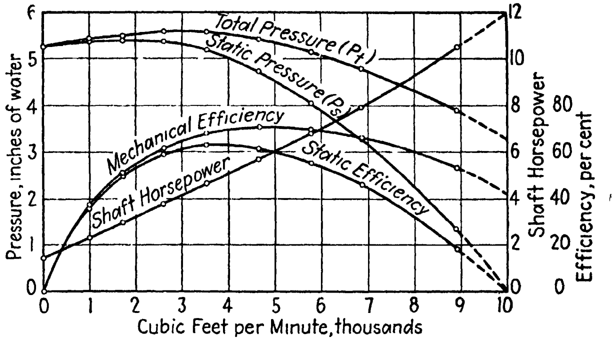


FIG. 178.—Typical fan-performance curves. (*From A.S.H.V.E. Test Code.*)

IV. NOTES

The fan test pipe should be smooth inside, and all joints should be airtight. A pipe area within 5 per cent of the area of the fan outlet is required by the Code.

The most common sources of error in fan tests are the following: (*a*) Zero of gages not accurately maintained. (*b*) Gages not accurately calibrated. The Code requires that all inclined gages be calibrated against a water-filled hook gage. (*c*) Inaccurate speed readings, or speed not accurately maintained. The Code requires three speed readings during each pitot traverse. (*d*) Air leakage. (*e*) Errors in determining horsepower, due to incorrect tare of the dynamometer or to excessive belt tightness or belt slippage if the fan is not direct driven. (*f*) Insufficient time taken to obtain a good average of gage readings in spite of momentary fluctuations.

Air volumes calculated from pitot-tube readings are usually higher than those obtained by nozzle or orifice measurement. Although the pitot values are standard, the A.S.M.E. Committee has recommended an alternate of nozzle or orifice readings, especially since differences of more than 3 per cent between the two methods are an indication of pulsating flow.

It is important that an adequate number of orifice sizes be used in Test 1. The A.S.H.V.E. Code calls for eight, but the A.S.M.E. Committee suggests four additional determinations in the range of volumes above the point of maximum efficiency, which is the important range of fan application.

TABLE XXVII.—TYPICAL PERFORMANCE OF CENTRIFUGAL AND PROPELLER FANS*

	Fan	Percentage of free-open volume			
		40%	60%	80%	100%
1. Total pressure, per cent of static no delivery (SND) pressure	*A*	103	96	83	77
	B	106	97	60	17
	C	56	39	31	14
	D	76	66	57	46
2. Static pressure, per cent of SND pressure	*A*	92	69	36	0
	B	103	91	50	0
	C	54	35	23	0
	D	68	48	27	0
3. Mechanical efficiency, per cent	*A*	72	60	47	39
	B	66	78	66	30
	C	28	38	40	18
	D	43	51	54	53
4. Horsepower, in per cent of maximum hp	*A*	29	48	77	100
	B	86	99	95	77
	C	73	61	51	45
	D	86	93	97	100

Fan *A*.	Forward curved blade centrifugal fan.
Fan *B*.	Backward curved blade centrifugal fan.
Fan *C*.	Axial flow fan.
Fan *D*.	Airplane propeller fan.

* From *Curves*, A.S.H.V.E. Guide.

For fans to be used without ducts, or for exhausters or for blowers with inlet boxes, special test duct arrangements are necessary, as given in the Code.

Ordinary tolerances recognize a possible variation of 2 to 3 per cent in duplicate tests to determine capacity, pressure, or horsepower.

A few typical fan-performance values are given in Table XXVII.

EXPERIMENT 69. Test of a Propeller Fan

I. PREFACE

Propeller or axial-flow fans have long been widely used for "free air" delivery, as desk fans, ceiling fans, man coolers and wall exhaust fans. Recent designs have also adapted them to duct operation against low or moderate pressures. Their main disadvantages as compared with centrifugal fans are higher tip speed and consequent noise, and usually lower efficiency. Their characteristic pressure and horsepower curves are not as well suited to most types of service as those of the centrifugal fan.

Advantages of simplicity, compactness, ease of installation, lower cost and direct motor drive frequently outweigh the disadvantages of the propeller fan, and aerodynamic studies have resulted in improved designs giving higher efficiencies and more desirable pressure characteristics. For operation at or near atmospheric pressure, however, the exact shape of the fan blade is much less important than might be expected.

Propeller fan tests are conducted in almost the same manner as centrifugal fan tests, and are covered by the same A.S.H.V.E. Standard Test Code. If the axial fan is for free-open inlet with duct discharge, the test setup is identical with that shown in Fig. 177, with the substitution of the propeller fan. The more common case of the propeller fan that is to be operated without a duct, requires a setup as shown in Fig. 179, with a still box having a minimum cross section at least ten times the outlet area of the fan under test. By varying the speed of the supply fan on the pitot-tube duct, any desired static pressure may be produced in the still box. Zero static or atmospheric pressure in the box is the condition most often required.

II. PREPARATIONS

The apparatus and setup, as already described, are essentially the same as for Experiment 68 (*q.v.*). The Code specifies an approach duct of such size as to give 1,800 fpm minimum duct velocity. If a Code test is not required, the tests may be abbreviated greatly by using interchangeable thin-plate metering orifices in the approach duct of Fig. 179, instead of making pitot-tube traverses.

Power measurements are usually made by calibrating the fan motor with a Prony brake (see typical curves of Fig. 81). Sometimes the over-all performance of fan and motor is reported, with no attempt to separate motor losses from fan losses.

III. INSTRUCTIONS

1. Test Procedure.—Take dimensions of fan, test pipe, still box, and orifices (if any). Observe zero reading of each manometer, and check zeros after each test run. Read barometer before and after tests. Accurate speed readings are essential, and for small fans the Strobotac or other accurate stroboscopic method is preferable.

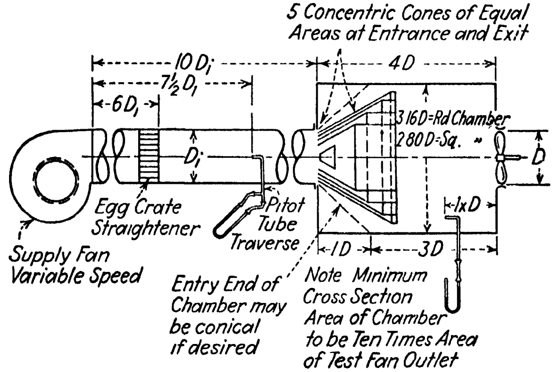

5 Concentric Cones of Equal Areas at Entrance and Exit
10 D_i
$7\frac{1}{2} D_1$
6 D_1
4D
3.16D=Rd Chamber
2.80D=Sq. "
D_i
D
Egg Crate Straightener
Pitot Tube Traverse
1xD
Supply Fan Variable Speed
1D
3D
Entry End of Chamber may be conical if desired
Note Minimum Cross Section Area of Chamber to be Ten Times Area of Test Fan Outlet

FIG. 179.—Test setup for axial-flow fans without duct. (*A.S.H.V.E. Test Code.*)

The fan motor should be given ample time to attain equilibrium temperature at each condition, before readings are taken. There should be no obstructions near the fan inlet or outlet when instruments are read. The fan should be accurately positioned in the shroud ring for each test, as this position has a major effect on fan capacity.

2. Determinations.—Several kinds of tests are possible: (*a*) A given fan operating at constant speed, against various static pressures. This is the usual Code test and the one most commonly required. The Code requires eight runs at approximately equal increments of fan capacity. (*b*) A given fan operating against zero static pressure (or any other specified pressure) and driven at various speeds. (*c*) Fan wheels of various designs, tested at the same tip speed or at the same static pressure or both. (*d*) Fan operating at the same speed and pressure, but with different positions of blades in the enclosing shroud ring or with various kinds of shroud rings. (*e*) Comparative tests by the A.S.H.V.E. Code method and the N.E.M.A. anemometer method (see Notes).

The exact test will depend on time and equipment available (see instructor for directions). Tests such as (*b*), (*c*), and (*d*) if required for zero pressure or free-open delivery only, should preferably be made in three runs, one at zero static, one at 0.03 to 0.05 in. positive and one at 0.03 to 0.05 in. negative static pressure.

3. Calculations and Report.—A suitable result sheet may be adapted from Form 68, since the items and the calculations are almost identical, except that items 11, 12, and 13 involving friction corrections are omitted when using the test setup of Fig. 179. In this case also, **the total pressure against which the fan operates** is the sum of the pressure read by the impact tube in the still box plus the equivalent velocity pressure produced by all the air flowing at a uniform velocity through the fan shroud ring. In other words, the velocity through the fan is $V = Q/A$, where Q is cubic feet per minute as measured and A is the area of a circle having a diameter equal to that of the fan ring.

Axial-flow fans follow approximately the fan laws of Table XXVI. If variable-speed tests are made, the results should be plotted on logarithmic coordinates as directed in Experiment 68. A Code test, at constant speed and variable pressure, will result in curves similar to Fig. 178 (see Table XXVII).

IV. NOTES

Anemometer tests for propeller fans are specified by the "**N.E.M.A. Fan Standards,**" issued by the National Electrical Manufacturers Association. For fans of small or medium size the anemometer is held in a plane 12 in. in front of the fan blades, and the readings are made by a traverse "at one inch intervals along horizontal and vertical axes." For large air-circulating fans an additional test is made "at 3 inch intervals along horizontal and vertical axes at a distance of 48 inches from the front edge of the fan blades to center line of anemometer." In either case "duration of readings shall not be less than one minute."

Two objections to this N.E.M.A. anemometer test are: (1) Entrained room air is measured, in addition to the air originally handled by the fan. (2) The instructions are not specific enough to ensure consistent results. The anemometer method has an advantage for small desk fans and other free-open fan wheels, in that no surrounding shroud ring is interposed during the test, as is required if the setup of Fig. 179 is used. The traverse at 48 in. distance gives a much higher reading than the traverse at 12 in. distance, and, since the N.E.M.A. method measures room air movement rather than fan capacity, the fan ratings obtained are of course much higher than those obtained by the A.S.H.V.E. Code tests.

EXPERIMENT 70. Test of Reciprocating Air Compressor

I. PREFACE

1. General.—The main purposes of an air-compressor test may be accomplished through the measurement of but three quantities:

a. The amount of air compressed and delivered per unit time.
b. The pressure difference between intake and discharge.
c. The power required by the machine.

The principal dimensions and the speed of the machine would of course be recorded also, together with certain data on the performance of the driving element (steam or gas engine, or electric motor). The taking of certain additional data may be required by the purpose of the test, as when the object relates to the fulfilment of a contract guarantee. But the accurate determination of even the three primary quantities mentioned above involves two of the most difficult of mechanical measurements, *i.e.*, quantity measurement of a compressible and noncondensable fluid, and the measurement of power input to a machine.

The measurement of the power required usually involves the application of an indicator to the compressor cylinders, and to the engine cylinders if the drive is a steam, gas, or oil engine. It may involve the use of a calibrated or cradle-mounted electric motor, or of a transmission dynamometer. The application and use of these devices have already been discussed, and the student should refer to these discussions.

A number of methods of air measurement are considered in Experiment 68, Test of Centrifugal Fan, page 334, and in Experiment 61, Flowmeters, page 296. Although the methods used in measuring air in a compressor test are essentially the same, the quantity of air to be measured is comparatively small, the pipe sizes small, and the pressure differences involved are large.

These facts eliminate the use of the anemometer and usually also that of the pitot tube and the venturi tube. The flow nozzle and the orifice are well adapted to these conditions, however, and these are the devices usually employed.

After the necessary measurements have been made, the performance of the machine must be expressed in such terms that the results shall be comparable with other compressor-test results. The recognized standards have been incorporated by the A.S.M.E. into a test code known as the "Test Code for Displacement Compressors and Blowers." This Code will be followed as far as practicable through the present test.

Since the thermodynamic behavior of air is very nearly that of a perfect gas, the performance of an air compressor furnishes one of the best possible examples of the direct application of the simpler laws of thermodynamics to the operation of an actual machine. The application of several familiar laws will therefore be recognized throughout the discussion that follows.

2. Methods and Results of Orifice and Nozzle Measurements.—The methods used for measuring the quantity of air handled by a compressor may be classified as follows:

a. Flow to atmosphere through an orifice or nozzle from a gaging tank on the compressor discharge.
  1. Using a "high-pressure" orifice of the sharp-edge type.
  2. Using a "high-pressure" rounded-entrance nozzle.
  3. Using a "low-pressure" orifice of the sharp-edge type.
  4. Using a "low-pressure" rounded-entrance nozzle.

b. "Flowmeter" (usually some form of orifice) inserted in the discharge line.

c. Flow through an orifice or nozzle into a gaging tank on the compressor intake.

d. Discharge into closed tanks of known capacity.

Method *d* (usually called a "pump-up test") is used chiefly for high-pressure multistage compressors discharging at 1,000 lb gage or higher.

Method *c* is used in testing vacuum pumps.

Method *b* is often the most convenient and practicable for tests of installed compressors of comparatively large size, such as those in use in boiler and railroad shops. The construction, installation, and use of the various types of flowmeters are fully covered in connection with Experiment 61, page 296.

Method *a*-4, using the low-pressure nozzle, is specified by the A.S.M.E. "Code for Displacement Compressors" (such as the reciprocating compressor being tested in this experiment). On the other hand, the A.S.M.E. Code on Instruments and Apparatus (Part 5, Chap. 4, par. 127) rates the low-pressure orifice as more accurate than the nozzle. Hence there is little choice between Methods *a*-3 and *a*-4, and a thin-plate orifice may in this experiment be substituted for the nozzle shown in Fig. 180.

Methods *a*-1 and *a*-2, using a small orifice or nozzle discharging directly from the high-pressure air line, require little or no special piping and are widely used for service tests, but they are not permitted under the Code.

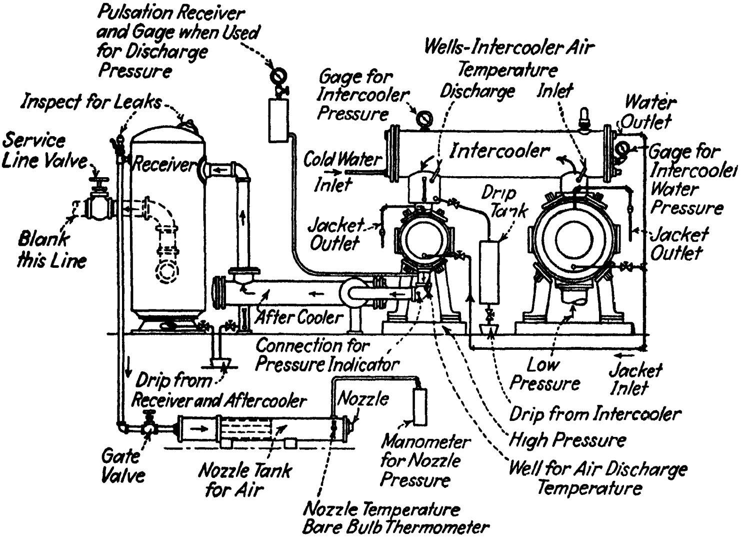

Pulsation Receiver and Gage when Used for Discharge Pressure
Inspect for Leaks
Service Line Valve
Blank this Line
Receiver
Gage for Intercooler Pressure
Wells-Intercooler Air Temperature
Discharge
Inlet
Water Outlet
Cold Water Inlet
Intercooler
Gage for Intercooler Water Pressure
Drip Tank
Jacket Outlet
Jacket Outlet
After Cooler
Connection for Pressure Indicator
Drip from Receiver and Aftercooler
Low Pressure
Jacket Inlet
Nozzle
Drip from Intercooler
High Pressure
Manometer for Nozzle Pressure
Gate Valve
Nozzle Tank for Air
Well for Air Discharge Temperature
Nozzle Temperature Bare Bulb Thermometer

FIG. 180.—Arrangement of gaging tank and other apparatus for compressor test. (*A.S.M.E. Code.*)

Method *a*-2, using the rounded-entrance nozzle is preferred, and the flow is calculated by Eq. (56), page 269.

II. PREPARATIONS

1. Instruments and Apparatus.

a. Gaging tank and nozzle or orifice for measuring air compressed.

b. Pressure gages, thermometers, and barometer.

c. Revolution counter.

d. Tanks and platform scales, calibrated tanks or water meters for measuring jacket water.

e. Indicators and planimeters.

f. Steam calorimeter, gages and thermometers, indicators, and means for condensing and weighing steam when driving element is a steam engine.

g. Indicators and means for measuring fuel when the driving element is an oil or gas engine.

h. Voltmeter, ammeter, wattmeter, and power-factor meter for measuring electrical input when the driving element is an electric motor.

2. Data to Be Taken.—Data or log sheets should be ruled in advance and should contain proper identification marks, including space for date and name of observer. Proper headings and subheadings should be given, including one column for "time" and space for "remarks." These suggestions apply to the records of data made by the individual observers, as well as to the main data sheet. The following measurements are necessary:

a. Diameter and stroke of compressor cylinders, diameter of piston rods.

b. Principal dimensions and complete rating of driving element.

c. Speed in revolutions per minute and total number of revolutions during test.

d. Indicated horsepower in compressor and in driving cylinders.

e. Intake, discharge, and barometric pressures.

f. Intake and discharge air temperatures, both wet and dry bulb.

g. Temperatures and quantity of cooling water.

h. Temperatures, pressures, and input of driving element.

i. Temperature and pressure of air entering orifice, and diameter of orifice (measured with micrometer).

III. INSTRUCTIONS

1. General Instructions.

a. All instruments should be calibrated or checked for accuracy.

b. The compressor should be operated under test conditions for a considerable period before observations are begun, in order to bring about a steady condition of pressure, temperature, and speed. A constant air temperature on the upstream side of the nozzle is usually an indication of steady conditions throughout.

c. Before starting the observations, all joints should be tested for leaks. This can best be done by applying strong soapsuds, with a soft brush.

d. The minimum duration of any compressor test must not be less than 1 hr (A.S.M.E. Test Code).

e. All conditions of operation must be kept as nearly uniform as possible during the test. (Particular care should be exercised to keep the "unloader" from operating.)

f. All readings should be taken simultaneously. It is particularly important that the other readings be taken at the instant the indicator cards are taken.

2. Results.—Form 70 is a suggested tabulation of test results. In the case of two-stage compressors, the following items will be duplicated, for the two cylinders: Nos. 9, 10, 11, 14, 18, and 22.

Form 70.—Results of Air-compressor Test

Object of test..
Owner............................Builder............................
Test by..........................Location..........................
Type of drive......................Method of volume control........
Cylinder diameter........Stroke.........Rated displacement......c.f.m.
Rated speed........................Barometric pressure..............

Item number	Item	Units	Run No. 1	Run No. 2
1	Duration of run....................	Min.		
2	Compressor discharge pressure, gage..	Lb. per sq. in.		
3	Temperature inlet air............	Deg. fahr.		
4	Temperature of discharge air... ..	Deg. fahr.		
5	Speed.................... .	R.p.m.		
6	Kind and size of orifice.... . . .			
7	Orifice pressure.............	In. water		
8	Orifice temperature......... ...	Deg. fahr.		
9	Indicated horsepower, air. . .	I. hp.		
10	Temperature cooling water in .	Deg. fahr.		
11	Temperature cooling water out ..	Deg. fahr.		
12	Weight of cooling water............	Lb. per min.		
13	Average piston speed...............	Ft. per min.		
14	Piston displacement per minute. .	Cu. ft.		
15	Free air delivered per minute	Cu. ft.		
16	Slippage.................	Per cent		
17	Isothermal air horsepower... .	Hp.		
18	Adiabatic air horsepower......	Hp.		
19	Gross horsepower.......	Hp.		
20	Volumetric efficiency...	Per cent		
21	Mechanical efficiency...............	Per cent		
22	Compression efficiency.......	Per cent		
23	Overall efficiency..................	Per cent		
24	Indicated horsepower in compressor cylinder per 100 cu. ft. of air delivered per minute at intake pressure and temperature.....................	Hp.		

Date............... Observers:.

The capacity of the compressor (item 15) is the quantity of air compressed per minute, expressed in terms of the volume at the temperature and pressure at the compressor intake. The conversion to intake conditions may then be made by using the gas equation $PV = wRT$. If moisture corrections are to be made, the methods given in the A.S.M.E. Code should be followed.

Gross horsepower is the indicated horspower in the steam or power cylinders in the case of a steam or internal-combustion-engine-driven compressor, or the brake horsepower delivered to the compressor shaft in the case of a power-driven compressor.

Adiabatic air horsepower is the horsepower required to compress adiabatically the measured quantity of air at the average rate from the average observed intake pressure and temperature to the average observed discharge pressure. For each cylinder of an air compressor the adiabatic horsepower is

$$\text{hp} = 0.0153 P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{0.283} - 1 \right] \tag{64}$$

where V_1 is the capacity at intake conditions, in cubic feet per minute, and P_1 and P_2 are pressures at intake and discharge, respectively, in pounds per square inch.

Isothermal air horsepower is the horsepower required to compress isothermally the measured quantity of air at the average rate from the average observed intake pressure and temperature to the average observed discharge pressure. This is the minimum power required by the ideal compressor, and it may be computed from the equation

$$\text{hp} = 0.00436 P_1 V_1 \log_e \frac{P_2}{P_1}. \tag{65}$$

Volumetric efficiency is the ratio of the capacity of the compressor to the displacement. The capacity is the actual amount of air or gas compressed and delivered, expressed in cubic feet per minute at intake temperature and pressure.

Mechanical efficiency is the ratio of the air indicated horsepower of the compressor cylinder to the gross horsepower.

Compression efficiency is the ratio of the horsepower required to compress *adiabatically* all the air delivered by the compressor to the horsepower developed in the compressor cylinder, as shown by the indicator cards.

Over-all efficiency is the ratio of the *isothermal* air horsepower to the gross horsepower.

After the complete results of the test have been calculated and tabulated, the efficiencies obtained should be compared with the values given in Table XXVIII and in the handbooks or textbooks covering the subject. A discussion covering these comparisons should be included in the report.

Table XXVIII.—Typical Efficiencies of Reciprocating Compressors

Class	Volumetric efficiency	Mechanical efficiency	Efficiency of compression
1. Small stationary air compressors, 3 to 10-in. bore, 250 to 800 rpm, double acting	65	85	85
2. Portable engine-driven air compressors, 3 to 8-in. bore, 500 to 1,500 rpm, single acting	60	75	85
3. Large two-stage air compressors, 6 to 18-in. bore, 200 to 500 rpm, double acting....	85	88	90
4. Commercial or small industrial refrigeration compressors, 2 to 5-in. bore, 300 to 1,000 rpm, multicylinder, single acting	65	60	65
5. Large ammonia or freon compressors, 5 to 12-in. bore, 200 to 800 rpm, double acting..	88	85	83

IV. NOTES

A pressure drop of 10 to 40 in. of water, across the nozzle or orifice, is preferred for accurate reading of a vertical water manometer.

Nozzles or orifices should not be placed directly in the intake or discharge pipes on account of pulsation errors.

The Code specifies that the discharge pressure gage shall be readable to 1 per cent of total scale, and of such range that the reading will lie between 50 and 75 per cent of the total scale.

Manometer tubing of at least 0.5 in. inside diameter is recommended by the Code, to make capillarity corrections unnecessary.

A positive-driven, totalizing counter is preferred for determining revolutions per minute, and this should run throughout the test.

For a two-stage compressor, the Code defines a compression efficiency for the entire unit on the basis of "minimum adiabatic horsepower." This requires additional calculations.

EXPERIMENT 71. Thermodynamic Analysis of Air-compressor Performance

I. PREFACE

The performance of an air compressor may be analyzed by the application of the ideal-gas laws (Table XX, page 235) and the general energy equation [Eq. (51), page 270], and a correlation thus obtained between the ideal and the actual machines. The exact type of analysis will depend on the compressor equipment available, but the best illustration is obtained through the use of a two-stage machine, with intercooler.

II. INSTRUCTIONS

1. Setup.—The compressor should be equipped with indicators on both cylinders, dynamometer or calibrated motor, thermometers at all locations for both air and cooling water, and an accurate air-metering unit.

2. Determinations.—Test procedure will be duplicated at each load, and the number of load runs will depend on the time available. Complete data for an energy balance should be taken, and all the necessary dimensions of the machine should be recorded, including the clearance volume at each end of each cylinder. Special attention should be paid to securing good indicator cards, and ample time must be allowed before each run for the establishing of "steady" conditions.

3. Results.—In addition to the complete energy balance, the following constructions should be made from the indicator cards: (1) Construct a "combined" indicator card, plotted to at least twice the scale of the low-pressure diagram, *i.e.*, lay out both diagrams to the same P-V scales on a single diagram. (2) Lay out on this same sheet the theoretical isothermal diagrams (see Fig. 58, page 88), with and without intercooling, starting with air at the intake conditions (or with the beginning of compression on the low-pressure card). (3) Plot on logarithmic paper the compression and expansion curves taken from a typical diagram for each cylinder, and determine from these lines the exponent in the equation of the polytropic curve (see Fig. 60, page 90).

Give a complete discussion of the heat-balance results and of the diagrams.

CHAPTER IX

STEAM POWER GENERATING UNITS AND AUXILIARIES

Introduction.—Recent developments in the power field have added many new features as to equipment and complications as to operation, through the introduction of ultra-high pressures and temperatures.

This chapter does not cover the testing of a modern plant as a complete unit or comparative tests of the various types of cycles when used singly or in combination.

The heading of the chapter designates that its treatment will be confined to a study of the elementary individual units of a steam power plant or station. An attempt is made to provide as far as possible in the engineering college laboratory the equipment and conditions found in practice, and, although the laboratory machines are usually small in size, the student should keep in mind that the laboratory tests are only incidental to the learning of the principles involved in this phase of engineering work, and that the observations and calculations on a given test are much the same for any size of machine.

It should also be recognized that it is not only the test engineer but also the plant superintendent or manager, the designer, the construction or erection engineer, the research or development engineer, and the operating man who are called upon to make or to interpret analyses involving the performance of the machines with which they have to deal and that such performance values are arrived at by tests of various kinds.

There are certain general preparations and preliminaries that should be completed before a test is started. There are also certain general instructions or principles that apply to the conducting of all tests and to the calculation and reporting of the results. Recognizing these facts, the A.S.M.E. Power Test Codes Committee has prepared a "Code on General Instructions," and this Code is to be used as a guide in all commercial work in power-plant testing. Special mention might be made of the following **general instructions:**

1. Ascertain the object of the test, and keep this object clearly in mind during the preparations for, and conduct of, the test.
2. Rule all data and log sheets in advance, and provide for at least one carbon copy. *All sheets should have at least one column in which time is recorded.* (Observations should never be designated by number only.) Each sheet should carry proper identification marks and titles and should be signed and dated by the observer.
3. Obtain and record the general descriptive data and dimensions and the manufacturer's rating and serial or identification number for each unit involved in the test. Examine and record the general features, arrangement, and condition of the apparatus and plant, and if need be make sketches to show the arrangement and any unusual features. When making such examinations, operating conditions bearing on the object of the test should be noted. So far as possible the examinations should be internal as well as external. All instruments and apparatus should be identified by numbers or otherwise, and this identification should appear on the log sheets.
4. Before starting the observations of each run, the apparatus should be run under test conditions for a sufficient length of time to bring about equilibrium conditions.
5. When possible, one or more preliminary tests should be made for the purpose of determining the adequacy of the instruments and for the training of the observers.
6. Every event connected with the progress of a test, however unimportant it may appear at the time, should be recorded on the proper log sheets, with the time of occurrence and the name of the observer who noted the event and entered the record.
7. When in commercial tests there are questions of fulfilment of contract involved, the parties should agree beforehand upon the operating conditions and test methods and upon the corrections to be made in case the conditions actually existing during the test differ from those specified in the contract.

One phase of the work so difficult for the student or even the engineer himself to understand is the time required for preliminaries. It has been said that the most inaccurate of all of an engineer's estimates is that of the time it will take him to do a test job. The operations of obtaining equipment, calibrating instruments, setting up, making a preliminary run and eliminating the difficulties shown by such a run usually consume a great deal more time than does the final test itself. In the laboratory, although an attempt is made to reduce to a minimum the time for preliminaries, unexpected difficulties often cut the test time down to an entirely insufficient amount. But, if the instructors and laboratory mechanics smoothed out all of the difficulties of setting up, the student would receive a one-sided training.

Test work in the steam power plant is of course chiefly concerned with the performance of the main units, *i.e.*, of the boilers and the engines or turbines. In connection with the performance of these units, however, certain other tests and analyses are desirable, or in some cases necessary. To complete a boiler test, for instance, it is necessary to obtain certain analyses of the fuel and also of the combustion products. In addition it might be desirable to determine the performance of certain of the boiler auxiliaries. In a condensing plant, the performance of the engine or turbine is to some extent dependent upon the performance of the condenser and its auxiliaries, and it is therefore desirable that the performance of these should also be measured. This point of view has determined the arrangement of material in this chapter. Not all power-plant auxiliaries are treated here, however, as some of them are also widely used in other fields outside the steam power plant, and these have been treated in other chapters. Specific tests or experiments dealing with power-plant equipment may therefore most easily be located by referring to the Index.

PART I. STEAM GENERATION—BOILER CONTROL, OPERATION, AND TESTING

PREFACE

In dealing with problems of control and operation in the boiler room, the engineer is faced with a many-sided problem. If he overlooks any one phase of this problem, he will meet with only partial success. Viewed from one angle, the control of the dual process of combustion and heat absorption is a scientific problem the successful solution of which depends on the understanding and proper application of certain familiar laws of physics and chemistry. To the practical fireman the control of the boiler fires and the maintenance of the boiler plant are largely a matter of skill and hard work—a trade learned by years of practice. The owner or manager of the plant considers the boiler room (or the entire power plant) in terms of the cost of its products, heat and power. He is not particularly concerned with either the science, the skill, or the labor involved in operating the boiler room, except as they may have some bearing on the cost of the product.

The technically trained engineer, especially if he has had some practical business experience, is in a position to appreciate each of these points of view. His technical training gives him an acquaintance with the scientific laws and a knowledge of how to apply them and some insight into the economic side of boiler-room management. At the same time, he learns much from the practical fireman and appreciates his skill in handling the boiler-room routine. Moreover, as an engineer he is accustomed to thinking in terms of efficiency and the ratio of output to input; and, since fuel costs are the largest item in boiler-plant costs, this state of mind gives him a good start on cost analysis. In large plants it is natural that the engineer should be recognized as the logical man to manage the boiler room.

Viewed from an engineering standpoint, boiler-room operation is composed of certain elements, each one of which is a typical engineering problem:

1. Materials handling, largely by mechanical means.
2. Control of the chemical process of combustion.
3. Control of the impurities in the boiler water.
4. Routine cleaning and inspection, and periodic repairs of heat-transfer equipment and accessories.
5. Handling and training of men for the accomplishment of these purposes.
6. Keeping of records showing plant performance and cost of operation.

These elements of boiler-room operation concern many engineers besides those directly charged with the operation of boiler plants. Someone must design these plants, someone must build them. In fact, every piece of equipment in the boiler room is the product of intensive development work on the part of engineers who have firsthand knowledge of boiler-room conditions. Large boiler rooms also involve complicated electrical layouts for power and control, since even the opening and closing of valves are done electrically. And the electrical load control and distribution for the entire plant, though primarily in the hands of the electrical operator at the main control board, are much more closely related to the boiler room than first thought would indicate. Electrical engineers, as well as mechanical engineers, are therefore closely concerned with what happens in the boiler room.

Returning to the six elements of boiler-room operation as listed above, it is seen that four of these are common engineering problems applied to a special type of work. That is, problems of materials handling, maintenance and repairs, handling of men, and determining performance and cost of operation are not peculiar to the boiler room. The control of combustion is on the other hand a technique that finds its most intensive application in the boiler plant. Nor is there any other field where the purification of natural waters has been given the attention it has received in the boiler plant.

It is to be noted that both of these problems (*i.e.*, control of combustion and control of impurities in the boiler water) call for chemical analyses of the products dealt with, and that these analyses must be made at frequent intervals if they are to be used as guides in operation. In other words, certain engineering tests and analyses must be made a part of the daily operating routine, if a boiler plant is to be operated efficiently. In this respect test work in the boiler room differs from tests on most other operating units. Performance data are usually valuable for cost analysis and distribution and are also of value to the designer but offer little of value to the operator of the machine. Once the design of an engine, a turbine, or a compressor is fixed, the operator can do little to affect its performance. Not so with a boiler installation. Its efficiency depends to a very large extent upon the operator.

Since the cost of fuel is by far the largest individual item of cost in a power plant, great care should be exercised in securing the greatest returns in the efficient use of the fuel.

It is common knowledge that the greatest savings to be made in the power plant are to be made in the boiler room, both from the proper control of combustion and in keeping of repairs to a minimum. In this the operator is greatly aided today by the use of modern instruments and automatic controls for the regulation of the combustion process. Boilers differing widely in design will give uniformly good results when properly operated. Thus it is that boiler control and operation are intimately related to boiler testing, and the student should keep this relationship in mind as we now proceed to take up the individual experiments or tests.

FUEL ANALYSIS

EXPERIMENT 72. The Proximate Analysis of Coal

I. PREFACE

"Fuel engineering" has recently come into recognition as a new specialty in the mechanical engineering field. Although the power-plant "fuel engineer" is perhaps the pioneer in this field, he is not by any means the only one who is intimately concerned with the economical utilization of fuels. In gas manufacture, in the metallurgical, ceramic, cement, and other industries, in the heating of buildings and in a large variety of industrial heating processes the study of the characteristics and the utilization of the available fuels is of prime importance. Usually, however, these fields do not call for the narrow specialization in fuel engineering which is justified in the steam central station.

The following facts serve to bring out the significance of the fuel problem in several of its aspects:

1. The United States uses approximately six times as much coal per capita as is used in Europe.
2. The United States uses over two-thirds of the world's production of petroleum.
3. The power consumption of the United States is nearly equivalent to that of the rest of the world.
4. Although many of our recently constructed steam power stations are maintaining average boiler efficiencies of 80 per cent, the average efficiency in the smaller plants is probably not over 60 per cent.
5. The supplies of practically all our high-grade fuels are limited, and prices are increasing.

These conditions are responsible for the following current developments: (1) focusing of attention upon the reduction of fuel costs in all heating processes; (2) attention to securing and maintaining high boiler efficiency in steam plants of all sizes; (3) development of methods for the economical utilization of low-grade coals; (4) progress toward increased thermal efficiency in the internal-combustion-engine field, this development centering largely around the diesel engine; (5) greater economy in the production of gaseous and liquid fuels from coal, and intensive experimentation looking toward a more general use of the coal-carbonization process. The major part in each of these lines of progress has of course been played by the research engineer.

II. COAL ANALYSES

The most important tests or analyses of a fuel are as follows:

Heating value, *i.e.*, the amount of heat (Btu per pound) that the fuel will furnish if burned under theoretically perfect conditions.

Ultimate analysis, a quantitative chemical analysis to determine the active chemical elements in the fuel (mainly carbon and hydrogen).

Proximate analysis of coal. Four constituents are distinguished in this analysis: (1) volatile matter; (2) fixed carbon or coke; (3) moisture; (4) ash.

Mechanical analysis of solid and pulverized coal. A determination of percentages of sizes by screen tests. The American Boiler Manufacturers Association has established the standard for pulverized coal that 65 per cent should pass a 200-mesh sieve.

Fusing point of ash, a test determined for solid fuels.

Flash point, viscosity, moisture, and sediment and other miscellaneous tests determined for liquid fuels.

What the Analysis Tells.—The *heating-value* determination forms the basis of all heat-efficiency calculations. It indicates the theoretical or "100 per cent efficiency" performance which is the ideal in all combustion processes. The *proximate analysis* gives a fair indication of the behavior of the coal in the furnace and also furnishes a basis for coal classification. The "volatile matter" is a good indicator of the length of flame that may be expected, the type of furnace that can be used most successfully, and the care that must be taken to secure smokeless combustion. "Moisture" is not only inactive, but it also absorbs heat and carries it up the stack, largely as latent heat of steam. The amount and appearance of the "fixed carbon" are a measure of the relative caking or coking properties of various coals. A large percentage of "ash" is undesirable, since it reduces the heat value of the fuel, increases the cost of transportation and handling, necessitates the disposal of large quantities of refuse, and often produces troublesome clinker.

From the *ultimate analysis* is calculated the air theoretically required to burn 1 lb of the coal, and in connection with the flue-gas analysis, the air actually used (see Experiments 75 and 78, pages 363 and 388). By far the largest loss in a steam boiler plant is that due to "excess air," and these determinations give a measure of this loss. By using Dulong's formula, the approximate heating value of the fuel may also be calculated from the ultimate analysis:

$$\text{Heating value} = 14{,}600\text{C} + 62{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 4{,}000\text{S}. \quad (66)$$

The *fusing point of ash* indicates the minimum temperature at which clinker will form. The latter in turn largely determines the type of stoking equipment and the rate of firing which can be used.

Coal Sampling.—The first step in coal analysis is the obtaining of a representative sample—and it is a very important step. A laboratory sample of 1 g must be representative of an original quantity amounting at times to several hundred tons, and the chance inclusion of a few pieces of slate or other impurities in a gross sample, which would otherwise have been representative, will cause the analysis and the tests in which the results are used to be in error accordingly. For this reason the size of the sample is dependent upon the size of lumps in the coal rather than upon the size of the original quantity being sampled.

The A.S.M.E. Test Codes recognize as standard the method of coal sampling that has been adopted by the American Society for Testing Materials, and this method is followed more or less closely in all commercial work. The main points involved in this standard method of sampling are presented in Fig. 181, from the Bureau of Mines *Bulletin.* Condensed instructions follow:

1. Coal is best sampled as it is being loaded, unloaded, or transferred.

2. A shovel is usually used, and increments of 5 to 10 lb in the small-sized coal and of 10 to 30 lb in the large-sized coal should be taken.

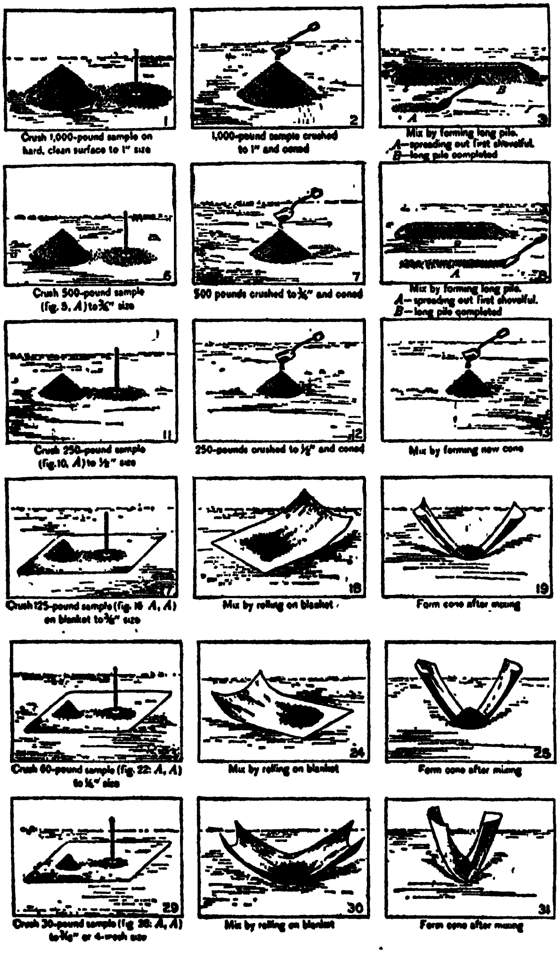
3. The increments should be taken regularly, at such intervals as will result in a gross sample of not less than 1,000 lb for original quantities of 500 tons or less. For small sizes of coal (largest pieces not over ¾ in.), a sample of 500 lb is sufficient.

4. The gross sample is crushed, mixed, and reduced in quantity to about 10 lb. This reduction is carried out in six stages:

a. Crush all large pieces in the 1,000-lb sample to 1-in. size or smaller, mix, form into a long pile (5 to 10 ft), and then halve by discarding alternate shovelfuls (Figs. 1 to 5).

b. Crush 500-lb sample to ¾-in. size, mix, and halve as before (Figs. 6 to 10).

c. Crush 250-lb sample to ½-in. size, mix by coning and reconing, form a cone-shaped pile, flatten top of same, divide into quarters, and discard opposite quarters (Figs. 11 to 16).



Crush 1,000-pound sample on hard, clean surface to 1" size

1,000-pound sample crushed to 1" and coned

Mix by forming long pile. *A*—spreading out first shovelful. *B*—long pile completed

Crush 500-pound sample (fig. 5, *A*) to ¾" size

500 pounds crushed to ¾" and coned

Mix by forming long pile. *A*—spreading out first shovelful. *B*—long pile completed

Crush 250-pound sample (fig. 10, *A*) to ½" size

250-pounds crushed to ½" and coned

Mix by forming new cone

Crush 125-pound sample (fig. 16 *A*, *A*) on blanket to ⅜" size

Mix by rolling on blanket

Form cone after mixing

Crush 60-pound sample (fig. 22: *A*, *A*) to ¼" size

Mix by rolling on blanket

Form cone after mixing

Crush 30-pound sample (fig. 28: *A*, *A*) to 3/16" or 4-mesh size

Mix by rolling on blanket

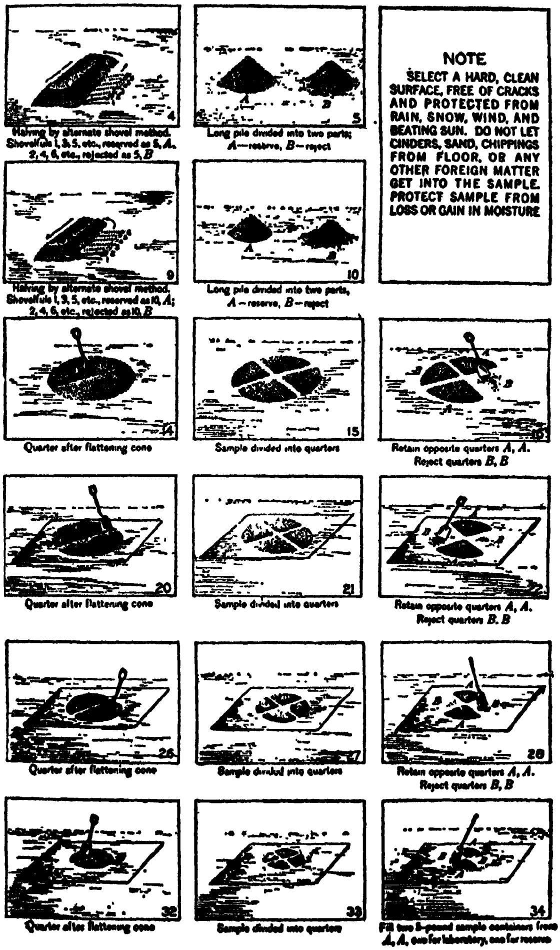
Form cone after mixing

Fig. 181.—Bureau of Mines

d. Crush 125-lb sample to ⅜-in. size, mix by rolling on a blanket, form cone, and again reduce by quartering and discarding opposite quarters (Figs. 17 to 22).

e. Crush 60-lb sample to ¼-in. size, mix on blanket, form cone, and reduce by quartering as before (Figs. 23 to 28).

f. Crush 30-lb sample to $\frac{3}{16}$-in. or 4-mesh size, mix, form cone, and quarter as before taking one or both of the opposite quarters for laboratory sample (Figs. 29 to 34).



Halving by alternate shovel method. Shovelfuls 1, 3, 5, etc., reserved as 5, *A*. 2, 4, 6, etc., rejected as 5, *B*

Long pile divided into two parts; *A*—reserve, *B*—reject

NOTE

SELECT A HARD, CLEAN SURFACE, FREE OF CRACKS AND PROTECTED FROM RAIN, SNOW, WIND, AND BEATING SUN. DO NOT LET CINDERS, SAND, CHIPPINGS FROM FLOOR, OR ANY OTHER FOREIGN MATTER GET INTO THE SAMPLE. PROTECT SAMPLE FROM LOSS OR GAIN IN MOISTURE

Halving by alternate shovel method. Shovelfuls 1, 3, 5, etc., reserved as 10, *A*; 2, 4, 6, etc., rejected as 10, *B*

Long pile divided into two parts, *A*—reserve, *B*—reject

Quarter after flattening cone

Sample divided into quarters

Retain opposite quarters *A*, *A*. Reject quarters *B*, *B*

Quarter after flattening cone

Sample divided into quarters

Retain opposite quarters *A*, *A*. Reject quarters *B*, *B*

Quarter after flattening cone

Sample divided into quarters

Retain opposite quarters *A*, *A*. Reject quarters *B*, *B*

Quarter after flattening cone

Sample divided into quarters

Fill two 5-pound sample containers from *A*, *A*, one for laboratory, one for reserve

method for coal sampling.

The laboratory sample must be kept in an airtight container. When the coal contains much moisture, a special moisture sample should be collected, in the quantity of about 100 lb. This sample is to be kept in a strictly airtight container until it can be reduced to a laboratory sample of about 5 lb. The reduction shall be made as quickly as possible and the laboratory sample placed in a tight container immediately after the reduction.

III. PREPARATIONS FOR THE PROXIMATE ANALYSIS

Apparatus.

1. Balance sensitive to 1/1,000 of the amount weighed. This will mean a chemical balance sensitive at least to $\frac{1}{10}$ mg. For the moisture determination a less sensitive balance is often used, and the weight of the sample will depend somewhat on the balance available.

2. Porcelain capsules (Royal Meissen No. 2) about $\frac{7}{8}$ in. deep and $1\frac{3}{4}$ in. in diameter, with flat aluminum covers.

3. Electric or other oven with temperature regulation to 225°F and having a circulation of air that has been dried by passing through concentrated sulphuric acid.

4. Gas or electric-muffle furnace with temperature regulation from 1300 to 1400°F.

5. Platinum crucible with tightly fitting cover, 10 to 20 cu cm.

6. Gas or electric tube furnace or muffle with temperature regulation to 1740°F (within 35°).

7. A porcelain crucible and a No. 5 Meker burner may be used in place of items 4, 5, and 6.

IV. INSTRUCTIONS

1. Procedure.—The procedure to be followed in the proximate analysis depends upon the purpose to be served by the analysis and the degree of accuracy required. Approximate determinations, especially those of moisture and ash, are often made in the boiler plant with such apparatus as is available. For a high degree of accuracy the apparatus and skill of a chemist must be employed, and in commercial boiler-test work the engineer seldom attempts to make his own fuel analyses. The large central stations, however, usually maintain laboratories well equipped for proximate and calorific analyses.

The American Chemical Society, the U. S. Bureau of Mines, and the American Society for Testing Materials have developed "official" methods for making each determination in the fuel analysis. These are the methods prescribed by the A.S.M.E. Power Test Code,[1] and they are described in the A.S.T.M. Specifications D22. In the following instructions, the first method given in each case is essentially the "official" method, while the second method may be used when time or apparatus is not available for the official method.

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[1] See "Test Code for Solid Fuels" (22 pp).

Preparation of Samples.—Laboratory samples should be prepared as outlined under Coal Sampling, above. It will be assumed in the following that the coal as received by the laboratory will pass a 4-mesh sieve, and in quantity amounts to 10 lb or less. If the sample contains much "surface moisture," that is, if it appears wet, it should be spread in a shallow pan, weighed, and air dried for 12 to 24 hr in a room or oven at a temperature below 100°F. This surface moisture should then be taken into account in calculating all subsequent values to the "as received" basis. The total laboratory sample should now be reduced to pass a 20-mesh sieve (a coffee mill is quite satisfactory for this reduction). A 60-g special moisture sample is to be taken immediately after the material has passed through the mill. This sample should be taken with a spoon from various parts of the 20-mesh product, placed directly in a rubber-stoppered bottle, and labeled "moisture sample, 20-mesh." The balance of the main sample should be thoroughly mixed, reduced on a riffle sampler, or by quartering, to about 120 g, ground to pass a 60-mesh sieve, transferred to a tightly stoppered bottle, and labeled "main sample, 60-mesh."

Moisture Determination.—Moisture must be determined for both the 20- and the 60-mesh samples. The whole purpose of taking the 20-mesh moisture sample is to determine the moisture loss, if any, during the grinding operation. Transfer with a spatula about 1 g of the 60-mesh sample to one of the porcelain capsules which has been previously heated and dried in a desiccator. Cover the capsule, weigh, then dry for 1 hr, uncovered, in the preheated oven at about 225°F. Cool in desiccator; weigh. The 20-mesh moisture sample should be similarly treated, using a 5-g portion and drying for 1½ hr.

Volatile Matter. *First Method.*—Weigh a 1-g sample of 60-mesh coal in a platinum crucible, cover, and place in furnace chamber which has been preheated to 1740°F. Heat exactly 7 min. Remove and cool without disturbing cover. Weigh. Loss of weight, minus moisture, equals volatile matter. Subbituminous and lignite coals should, on account of their high volatile content, be subjected to a preliminary slow heating over a bunsen flame for 5 min.

Second Method.—This method is only fairly accurate, because the porcelain crucible retards the transmission of heat to the sample and because the Meker burner has a rather uncertain heating capacity. A No. 5 Meker burner is used, and above the burner is mounted a nichrome triangle, on which is placed an inverted 20-g assay crucible with the bottom ground off, exposing a hole about 1 in. in diameter. After this apparatus has been preheated to as high a temperature as possible, place a porcelain crucible containing a 1-g sample of air-dried coal under the assay crucible. Ignite for 7 min as in first method (highly volatile coals should be preheated). Without disturbing cover, transfer crucible to desiccator after it has cooled below redness, and weigh when cool. Loss of weight, minus moisture, equals volatile matter.

Ash.—The ash determination may be made either on the 60-mesh (dried) moisture sample or on the residue from the volatile determination.

First Method.—Place the uncovered weighed porcelain capsule containing coal (or residue) in a muffle furnace and gradually heat to redness. Finish ignition at about 1350°F, with occasional stirring, until all carbon particles have disappeared. Cool in desiccator and weigh. Then continue alternate heatings and weighings until weight is constant.

Second Method.—This is similar to the first method except that a large Meker burner is used, if muffle is not available. If the moisture sample is used for this determination, it must be subjected to preliminary heating for about 15 min to expel the volatile matter. This heating should be carried out at a sufficient distance above the flame so that there will be no mechanical loss of solid particles owing to the rapid expulsion of volatile matter. Ignition is completed by applying the full blast flame to the bottom of the capsule or ashing dish. This determination will probably require the greater part of an hour for completion. Be sure to ignite to constant weight.

2. Determinations.—All determinations are to be made in duplicate and should check each other within 1 per cent.

Form 72.—Report on Coal Analysis
(Proximate analysis and heating value)

Field Record:
Coal lot designation or No.................................Quantity.......
Size (original)........Sampled by....... ....Date........
Source of coal: State.................County, field, district, or trade name...
Laboratory Record:
Sample designation and No...
Date received................Quantity..............Fineness........
Analyzed by.................................Dates............
Results of Analysis:
Coal classification...
Coking or non-coking...
Appearance of ash..

Proximate analysis	As received	Dry coal
Moisture, percentage.....................		
Volatile matter, percentage...............		
Ash, percentage.........................		
Fixed carbon, percentage................		
Calorific analysis: B.t.u. per lb...........................		

Analysis reported by
....

3. Results and Calculations.—The percentage of "fixed carbon" equals 100, minus the sum of the percentages of moisture, volatile matter, and ash. All results should be calculated both on the "as received" and on the "dry" basis. Especial care must be exercised in making the various corrections for moisture. It should be remembered that the coal "as received" includes the surface moisture and the moisture lost during grinding, as well as the moisture found in the 60-mesh sample. The report on this experiment should give all calculations in detail and also a tabular summary of results. A discussion should be written giving the classification of the coal and also giving conclusions regarding the probable behavior of this coal in a boiler furnace.

V. PRECAUTIONS

1. Label all containers used, including the porcelain capsules.
2. Record *all* weights in notebook or on data sheet, giving, in each case, full data as to what they represent.
3. Do not throw away any coal or residue until you are *certain* that it will not be needed for a check determination, or for another analysis.

EXPERIMENT 73. Calorific Analysis of Coal

I. PREFACE

Almost every branch of mechanical engineering is to some extent concerned with the production of heat or power. The equipment under consideration may be an airplane engine or a steam central station, it may be a factory power plant or a home heating plant. But in any case, the engineer's study of efficiency begins with a consideration of the characteristics of the fuel used. The most important of these characteristics is the "heating value," or total amount of heat which is *available* in a unit quantity of fuel. This amount forms the denominator in the efficiency fraction:

$$\text{Efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{heat utilized}}{\text{heat available}}.$$

A **fuel calorimeter** is a device in which a fuel sample may be completely burned while the heat it produces is absorbed and measured. Such calorimeters fall into two general classes, the "continuous" and the "discontinuous" types. In the continuous calorimeter both the fuel (liquid or gas) and the heat-

Form 73.—Heating Value of Coal by Oxygen-bomb Calorimeter

Coal sample designation and number...... ....Date.. ....

Moisture content of coal....per cent Observers......................

Item number	Item		Observations and results	
1	Initial weight of flask and fuel . ..			
2	Final weight of flask and fuel			
3	Net weight of fuel burned			
4	Weight of water in calorimeter			
5	Calorimeter water temperatures, deg. centigrade, time in minutes			
	0	8.0		
	1	8.5		
	2	9.0		
	3	9.5		
	4	10		
	5 (fired)	11		
	5.5	12		
	6.0	13		
	6.5	14		
	7.0	15		
	7.5			
6	Jacket-water temperature, deg. centigrade........			
7	Length of wire burned, in.......................			
8	Temperature at firing, observed..................			
9	Maximum temperature reached, observed.........			
10	Deg. rise per min. before firing..................			
11	Deg. fall per min. after maximum temperature.....			
12	Time from firing to 60 per cent maximum temperature..			
13	Time from 60 per cent maximum to maximum temperature...................................			
14	Temperature at firing, corrected.................			
15	Maximum temperature reached, corrected.........			
16	Corrected temperature rise......................			
17	Temperature rise per g. of coal.............. ...			
18	Gram-calories per g. of coal......................			
19	B.t.u. per lb. of coal...........................			

absorbing medium (water) flow through the calorimeter at measured rates and the entire process is a continuous one (see Experiment 74, page 359). In the discontinuous type, which must be used for solid fuels and is often used for liquid fuels, a weighed sample is burned and the heat thus generated is absorbed by a known weight of water.

Of the various discontinuous calorimeters that have been devised there are only two kinds that are used to any extent in commercial determinations of the heating value of coal. These are the "oxygen-bomb" and the "peroxide-bomb" calorimeters. The bomb in either case forms a heavy-walled steel combustion chamber provided with an electric ignition apparatus for firing the coal sample. In the oxygen-bomb calorimeter a supply of pure oxygen gas is forced into the bomb to a pressure of 200 to 500 lb per sq in. In the peroxide-bomb (Parr) calorimeter a chemical, sodium peroxide, is mixed with the coal, and this liberates the necessary oxygen.

The oxygen-bomb calorimeter is conceded to give the most accurate results, but the Parr peroxide-bomb calorimeter is widely used in commercial work because of its cheapness and because of the ease and rapidity with which the determinations may be made.

In the present experiment, the oxygen-bomb calorimeter is to be used. The instructions given are general, but the Mahler, Emerson, and Parr bomb calorimeters have been kept in mind particularly.

For a complete discussion of the approved methods for calorimetric determinations, consult the A.S.M.E. Code on "Heat of Combustion," Instruments and Apparatus, Part 9.

II. PREPARATIONS

1. Apparatus.—The following are required: Oxygen-bomb calorimeter consisting of bomb with fuel pan and wrench, water can, jacket, ignition device, and stirrer. Oxygen cylinder with pressure gage and connections. Scales for weighing water, and chemical balance for weighing the coal sample. Stop watch. Calibrated thermometer which can be read to 0.01°C or less. (Thermometers furnished with bomb calorimeters are usually calibrated in degrees centigrade.)

III. INSTRUCTIONS

1. Procedure.—Use part of the 60-mesh air-dried coal sample prepared for the proximate analysis. It is more convenient to use a weighing vial or flask rather than to weigh the sample in the calorimeter fuel pan. Pour about 5 g of the coal sample into the weighing flask, cork same, and weigh to 0.1 mg. Place calorimeter fuel pan upon its support and attach ignition wire, making it dip in the pan without touching it. Pour from the weighing flask into the pan approximately 1 g of the coal, then reweigh the flask and determine by difference the amount of coal used. After making sure that the ignition wire is touching the coal, close the bomb, tightening same with special large wrench. *Without tipping or jarring,* connect bomb to oxygen tank and fill slowly to a pressure of about 300 lb. Close valves, detach bomb, and immerse in water to detect leaks, if any. Wipe dry. In the calorimeter water can, weigh out 1,900 g of water (some calorimeters call for 2,000 g) at a temperature of 2 to 4°F below the temperature of the room (or of the water in the calorimeter jacket if a water-jacketed calorimeter is used). Place can in jacket, make electrical connections to bomb, and lower same carefully into the water can. Put on calorimeter cover, and lower stirrer and thermometer into their respective positions in the water can. The thermometer should be immersed about 3 in. and should stand at least ½ in. from the bomb. Start stirrer.

The Run.—When temperatures have become equalized, start taking temperature readings, one each minute, reading the thermometer as closely as possible, while an assistant calls time and records the readings. At the end of 5 min, close the ignition circuit for 1 or not more than 2 sec, and continue readings at ½-min intervals for 5 min, then at 1-min intervals for 5 min, making 15 min in all (see data sheet). Carefully remove thermometer and take a reading of the jacket-water temperature with the same, then place the thermometer in its protecting case. Remove the bomb, and, *after* releasing the pressure, open the same and examine residue. Also examine interior of bomb for soot deposits, and, if any are found, repeat the determination. Wipe lining with a dry cloth before putting bomb away.

2. Determinations.—Heating-value determinations should always be made in duplicate, with a reasonable check.

3. Results and Calculations.—The sequence of calculations is indicated on the data sheet, but certain items call for explanation. It is necessary to modify the temperature readings slightly in order to correct for radiation, and there are several methods of doing this. Dickinson's method will be used, as it is simple and fairly accurate and is the one recommended by the A.S.T.M.

The procedure for this method of correction is as follows: Correct all thermometer readings for thermometer calibration. (Result sheet, Form

73, should show corrected temperatures only.) Plot a time-temperature curve for all readings. The rise in temperature before firing and the fall in temperature after the maximum has been reached should plot as straight lines. From the slope of these lines, determine the rates of rise and fall, respectively, for these intervals. Locate on the graph the point at which the total temperature rise reached 60 per cent of its total amount. Then:

Temperature at firing as corrected equals the observed temperature at firing plus the product of the rate of temperature rise before firing and the time required to reach 60 per cent of maximum temperature, as determined from the curve.

Maximum temperature, as corrected, equals the observed maximum temperature plus the product of the rate of temperature fall after maximum temperature and the time elapsed between 60 per cent of maximum temperature and maximum temperature, as determined from the curve.

In terms of the item numbers used on the data and result sheet, this would be:

$$\text{Item } 14 = \text{item } 8 + (\text{item } 10)(\text{item } 12)$$
$$\text{Item } 15 = \text{item } 9 + (\text{item } 11)(\text{item } 13)$$

Further corrections may be applied if a high degree of accuracy is required. These corrections are for the heat given off by the ignition wire and for the formation of nitric and sulphuric acids during combustion. These refinements will be omitted in this case, however.

EXPERIMENT 74. Calorific Analysis of Gas or Oil Fuel

I. PREFACE

For gaseous fuels and for light liquid fuels, such as gasoline, kerosene, benzol, or alcohol, the most convenient and the most common apparatus for the determination of heating value is a continuous calorimeter of the Junkers type. The oxygen-bomb or the Parr peroxide calorimeter may be used for light liquid fuels if proper precautions are taken, but it is somewhat difficult to obtain complete combustion in these types of calorimeters.

In the Junkers calorimeter (Fig. 182), the combustion space and the flue-gas passages are surrounded by a suitable water jacket. Water is kept flowing through this jacket at such a rate that the products of combustion leave the apparatus at the temperature of the entering fuel and air. All the heat generated by the fuel under test is thus absorbed by the water. Once this condition is established, the process is a continuous one, since the fuel and the cooling water are supplied at constant pressure. Runs of any convenient duration may then be made.

For gases a bunsen burner is used, with a wet gas meter of the positive-displacement type for measuring the fuel. Liquid fuels are burned in a special burner of the generating or blowtorch type, and, for determining the quantity of fuel burned, the complete blowtorch is suspended from a sensitive balance.

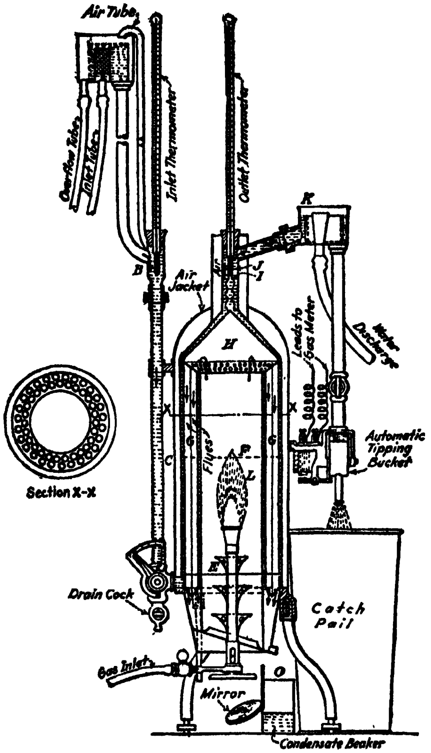

FIG. 182.—Sargent gas calorimeter (Junkers type).

The water resulting from the burning of the hydrogen constituent of the fuel is condensed, collected, and measured.

The heating value as determined directly from calorimeter readings is called the "**higher heating value.**" In furnaces and most other types of combustion apparatus the heat absorbed by the water formed from the burning of hydrogen is not recoverable (as it is in the calorimeter), because the products of combustion escape at a temperature above that of the vaporizing point of water, or more accurately, above the dew point of the steam-air mixture. The **"lower heating value"** is therefore often desired, and it is obtained by subtracting this "hydrogen loss" from the higher heating value. Strictly speaking, the products of combustion in a flue or exhaust pipe exist in the form of a steam-air mixture, and Dalton's aw of partial pressures may be applied to the mixture. It becomes necessary, if strict accuracy is desired, to determine the partial pressure due to the steam and to obtain from the steam tables its properties at this pressure.

The A.S.M.E. Code specifies that for obtaining the lower heating value (*LHV*) a latent heat of 1040 Btu per pound be used, *i.e.*:

$$LHV = HHV - 1040 \times 9 \times H$$

where H is the hydrogen in the fuel expressed as a decimal.

It is common practice to calculate the calorific value of petroleum fuels by use of the Sherman and Kropff equations, as given on page 436.

These formulas give a close approximation for distilled fuels. Their use is often justified by the experimental difficulty of the calorific determination with a volatile fuel.

II. PREPARATIONS

1. Apparatus.—The apparatus required is as follows: Junkers calorimeter with bunsen burner and gas meter for gas fuel or with blowtorch and special sensitive balance for liquid fuel. Rubber hose for supply, discharge, and overflow connections. Two special precision thermometers for water inlet and outlet. Three 120°F (or 220°F) thermometers for room temperature, flue temperature, and gas temperature. Sensitive platform scales and two water buckets for weighing jacket water; 100-cu-cm graduate for measuring water formed by burning of hydrogen.

2. Setup.—Attach rubber hose for supply, discharge, and overflow. Place all thermometers. Place bucket under water outlet, open water tap carefully, and turn inlet valve on calorimeter to regulate flow. *When satisfactory water circulation has been secured,* light and insert burner, making sure that burner flame is exactly in the center of the combustion chamber. (See instructor for special directions for handling liquid-fuel blowtorch.)

III. INSTRUCTIONS

1. Procedure.—Regulate the water flow and the burner flame so that the products of combustion are discharged at approximately room temperature. When all temperatures become steady, the run may be started. Temperature readings and gas pressure are to be taken every minute, and water and fuel quantities for the entire run. Read barometer.

2. Determinations.—For gas fuel, make three runs, burning 60 liters of gas for each run. For liquid fuel make three runs, burning 10 g of fuel for each run. If runs do not check each other closely, additional runs should be made.

3. Results and Calculations.—Inasmuch as it is most convenient to use English units (pounds) in measuring the cooling water, and metric units in measuring the fuel and the condensate, care must be taken not to confuse the two systems in calculating the results. The heating value of the fuel is to be expressed in Btu per pound, and both higher and lower heating values are to be reported (see page 360). For gas fuel these values are to be expressed both at the measured pressure and temperature, and at standard conditions of 14.7 lb abs and 32°F. This conversion is accomplished by using the gas law $PV = wRT$.

As a further refinement in the determination of heating value, a correction may be made for the heat carried away by the moisture in the exhaust gases. If the relative humidity of the air in the test room is as low as 20 per cent and the temperature is high, the correction will amount to about 10 Btu.

IV. PRECAUTIONS

1. Be sure that water is circulating through the calorimeter before introducing flame, as all joints are soldered.
2. Handle precision thermometers with care. Watch outlet temperature carefully, as thermometer is easily broken by excessive water temperature.
3. Adjustments may be made between runs, but do not disturb valve settings during a run unless to protect precision thermometer.

EXPERIMENT 75. The Analysis of Flue Gases or Products of Combustion

I. OBJECT

This experiment is intended to familiarize the student with the technique of making gas analyses with the Orsat apparatus and with the use of these analyses in calculating the weight of combustion products per pound of fuel burned.

II. PREFACE

There are five principal constituents usually found in flue or exhaust gas, *viz.*, carbon dioxide, oxygen, carbon monoxide, nitrogen, and water vapor. Under some conditions hydrogen and hydrocarbons may be present in important quantities, and there may also be present traces of sulphur dioxide and trioxide. Usually the five main constituents give a sufficiently accurate indication of furnace conditions. The carbon dioxide content alone gives a rough approximation of these conditions.

The Orsat is the standard apparatus for the analysis of flue gases from furnaces and of exhaust gases from internal-combustion engines. In large power plants, recording CO_2 machines and steam-flow air-flow meters may be used, but in setting and calibrating these instruments the engineer again resorts to the Orsat apparatus.

The Orsat analysis is made by successively absorbing CO_2, O_2, and CO from the flue-gas sample. The remainder is reported as nitrogen. From this analysis may be calculated the weight of the flue gases per pound of carbon burned, and, if the fuel analysis and stack-gas temperature are available also, all the stack-gas heat losses may be calculated (see page 389). The stack losses include the sensible heat of the dry stack gases, the losses due to moisture in the stack gases, and the loss due to unburned carbon monoxide. (In rare cases there may be a loss due to unburned hydrocarbons, and this cannot be determined from the Orsat analysis.)

III. PREPARATIONS

1. Apparatus.—The following are required: (1) Orsat apparatus; (2) storage tank for gas samples; (3) gas pump or aspirator;

(4) rubber tubing and connectors; (5) gas-sampling pipe: (6) data boards.

The Orsat should be set at a convenient height, in a good light and in a location where it is not likely to be overturned and broken.

The open end of the sampling pipe should be placed in what the observer judges to be the center of the gas stream (see page 392).

The connections should be so arranged that one operator may be drawing a fresh sample while another completes the analysis on the previous one.

If other types of gas analyzers are available, they should be connected so that simultaneous readings from each may be compared with the Orsat readings.

IV. INSTRUCTIONS

1. Procedure.—The work is most conveniently carried out if one man draws the gas samples, a second operates the Orsat, and a third records the data.

The data should include the time of taking each sample, the condition of the fire at that time, and simultaneous readings of stack temperature, steam flow, and draft if these are available. Certain general data are also required (see Form 75).

The construction of the Orsat should first be studied (Figs. 183 and 184). It consists of three main parts. (*a*) A measuring burette *A*, usually water-jacketed to maintain constant temperature. (*b*) A "leveling bottle" *F*, partly filled with water and connected to the burette by a rubber hose. By the use of this bottle the gas may be drawn into or forced out of the burette. (*c*) A series of three absorption pipettes *B*, *C*, and *D*. These are used in order, beginning next to the burette, and they absorb CO_2, O_2, and CO, respectively.

Before attempting to use the Orsat apparatus, the student should be thoroughly familiar with the Notes and Precautions given in Sec. V, page 368. The manipulation for the analysis consists of the following steps:

a. Flush out the burette once or twice with fresh gas and then draw in slightly more than 100 cu cm of gas.

b. Bring the burette water level exactly to the zero mark by discharging the excess gas to the atmosphere, taking care not to introduce any air. Read the water level from the bottom of the meniscus.

c. Holding the leveling bottle well above the zero mark, open the stopcock to the first pipette and pass all the gas slowly in and out of this pipette several times. Bring the reagent in the pipette back to the zero mark in the capillary neck before closing the stopcock.

d. Read and record the new water level in the burette; this will show the percentage of CO_2 which the reagent has absorbed.

e. Repeat operations *c* and *d* with the **same** pipette until further manipulations do not increase the burette readings, *i.e.*, until two successive readings check each other.

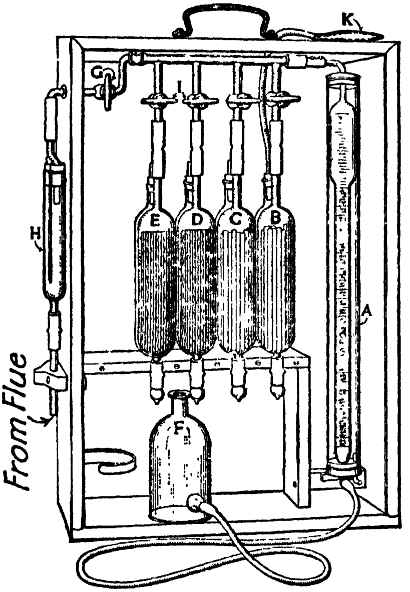

FIG. 183.—Orsat apparatus with two carbon monoxide pipettes.

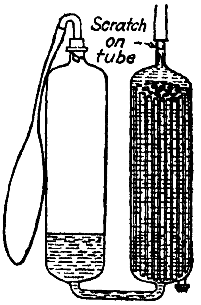

FIG. 184.—Detail of Orsat pipette.

f. Repeat operations *c*, *d*, and *e*, using the second, or oxygen, pipette. The final water level in the burette will represent the total of CO_2 and O_2, and the CO_2 percentage must be subtracted in order to determine oxygen.

g. Repeat operations *c*, *d*, and *e* for the third, or CO, pipette.

2. Determinations.—Each observer should make several determinations as directed by the instructor. Each observer is to obtain a copy of all the data taken.

3. Results and Calculations.—The data and results are to be reported in the form of a tabulation showing, for each analysis made during the period:

a. The analysis of the flue gas by volume, in terms of percentage of CO_2, of O_2, of CO, and of N_2.

b. The corresponding analyses by weight.

c. The sum of CO_2 plus O_2 plus $\frac{1}{2}$CO, for each analysis.

d. The condition of the fire and other miscellaneous data for each sample.

e. The calculated weight of flue gases per pound of carbon burned, for each sample analyzed. (A suggested form of data and result sheet is given herewith.)

The sum referred to in item *c* above merely furnishes a rough check on the accuracy of the analysis and does not enter into the computation of gas quantities or heat losses. This check is based primarily on Avogadro's law, which states in effect that a molecule of any gas occupies a certain volume and that this volume is the same for any and all gas molecules. If, then, a molecule of oxygen unites with an atom of carbon to form a

Form 75.—Results of Flue-gas Analysis

Location of plant.......................Owners..........................
Kind of boiler............................Size of boiler...............
Type of setting.........................Method of firing................
Method of sampling gases...

Item number	Item	Run No.				
		1	2	3	4	5
1	Time........................					
2	Condition of fire..............					
3	Steam flow, lb. per hr...........					
4	Draft below damper, in.........					
5	Draft over fire, in..............					
6	Flue-gas temperature...........					
7	CO_2 by volume................					
8	O_2 by volume.................					
9	CO by volume................					
10	N_2 by volume................					
11	$CO_2 + O_2 + \frac{1}{2}CO$............					
12	CO_2 by weight................					
13	O_2 by weight.................					
14	CO by weight.................					
15	N_2 by weight.................					
16	Pounds of flue gases per lb. of carbon burned................					

Date................ Observers:..
....
....

molecule of CO_2, the resulting product will occupy the same volume as the original oxygen. But if a molecule of oxygen unites with two atoms of carbon to form two molecules of CO, the resulting product will occupy twice the volume of the original oxygen. For a fuel composed of carbon only, therefore, the sum of the percentages by volume in the flue gas, of CO_2 plus O_2 plus $\frac{1}{2}CO$, should equal the percentage by volume of oxygen in the original air, or 21 per cent. Since almost all fuels contain some hydrogen, the actual sum is always less than 21 per cent as the H_2O disappears (condenses) when the gas sample is cooled, and the accompanying nitrogen remains to dilute the flue gas. The theoretical value for this sum may be calculated for any given case, the limiting values, *for perfect combustion*, being about 18 per cent for bituminous coal, 15 per cent for oil, and 11.7 per cent for natural gas. In actual cases, the sum will be slightly less. (Figure 192 gives values for oil fuel.)

The calculations necessary to determine item *e*, the weight of flue gases per pound of carbon for each sample, are based on the combining weights of carbon and oxygen. But since the flue-gas analysis is volumetric, it is first necessary to reduce to analysis by weight, as required in item *b*. This again involves Avogadro's law and the molecular weights of the flue-gas constituents. Having ascertained the flue-gas analysis by weight, the carbon can be separated out and the pounds of carbon per pound of dry flue gas thus determined. The weight of carbon is of course equal to $^{12}\!/_{44}$ the weight of CO_2 plus $^{12}\!/_{28}$ the weight of CO. We wish to obtain the pounds of flue gas per pound of carbon burned, and this is evidently the reciprocal of the pounds of carbon per pound of flue gas. If the fuel is not pure carbon, the calculation becomes:

$$\begin{aligned}\frac{\text{Pounds flue gas}}{\text{Per pound fuel}} &= \frac{\text{pounds carbon}}{\text{per pound fuel}} \div \frac{\text{pounds carbon}}{\text{per pound flue gas}} \\ &= \frac{\text{pounds carbon}}{\text{pounds fuel}} \times \frac{\text{pounds flue gas}}{\text{pounds carbon}}. \qquad (67)\end{aligned}$$

If the student will follow the above calculation analysis step by step, using numerical values from his actual flue-gas analyses, he should have no difficulty in computing the values called for.

For the computation of flue-gas heat losses and of total air supplied additional information is necessary, particularly flue-gas temperature and certain items of the fuel analysis. These calculations are therefore taken up in Experiment 78, Boiler Test, page 387.

V. NOTES AND PRECAUTIONS

It is difficult to obtain a high degree of accuracy in flue-gas analyses, and it is particularly important that all reasonable precautions be taken to prevent errors and inaccuracies. The following suggestions cover most of the common difficulties:

1. The reagent in each of the pipettes must be drawn up to the zero mark in the capillary before the gas sample is taken in.
2. To obtain atmospheric pressure in the measuring burette when a reading is being made, the leveling bottle must be held so that its water level corresponds with the water level in the burette.
3. The apparatus should be carefully tested for leakage before an analysis is started. A simple method of doing this is to take in a 100 cu-cm sample of gas (or air), and then set the leveling bottle on the top of the frame. Leakage at the stopcocks or connections of the absorption pipettes will cause the reagent to recede from the zero level. Leakage in the balance of the apparatus may be detected by again checking the zero in the measuring burette by bringing the water levels in line. If leakage is indicated, but the source of trouble cannot be located, a coating of thick shellac applied at all joints will usually stop the leakage. The apparatus should be under a slight vacuum while the shellac is being applied.
4. The burette should be flushed once or twice with "fresh" gas before a sample is taken for analysis.
5. The analysis must always be carried out in the proper sequence, *i.e.*, CO_2, then O_2, then CO, as the oxygen solution will also absorb carbon dioxide, and the carbon-monoxide solution will absorb oxygen. For this reason it is also very important that the absorption be carried to completion in each pipette before admitting the gas into the following one.
6. When there is any doubt about the strength of the solutions they should be renewed. The CO_2 solution will absorb about forty times its own volume of gas, while the other solutions will absorb only one or two times their own volume. The oxygen solution is always exhausted first, and a more highly absorbent solution may be made by preparing two reagents and mixing them at the time of use. *Solution* 1 is 34 g of dry caustic in 85 g of water. *Solution* 2 is 40 g of pyro in 80 g of water. A recent extensive investigation on Orsat reagents selected this as the best oxygen reagent and reported that it "removed all oxygen from air in six passes when fresh and in seven passes after thirty times." Air should be excluded from both the oxygen and the CO reagents from the instant they are mixed. Light also has a deteriorating effect on the oxygen solution.
7. The water in the burette and leveling bottle should be saturated with gas before a sample is taken in for analysis.

8. Care must be used not to allow either the burette water or the reagents to enter the capillary manifold above the stopcocks. If the solutions are allowed to mix in this manner, they must be renewed before further analyses can be made.

9. Owing to the presence of hydrocarbons in natural gas, an analysis for CO_2 and CO is insufficient to determine incomplete combustion.

10. The first pipette, for absorbing carbon dioxide, contains a solution of potassium (or sodium) hydroxide in the proportions of 2 parts by weight of water to 1 part of the caustic sticks (2 pt to 1 lb). The second pipette, for absorbing oxygen, contains potassium pyrogallate. An easy way to prepare the solution is to put 5 g of the dry (resublimed) acid into a funnel, insert the funnel in the mouth of the pipette, and pour over the acid a sufficient amount of caustic potash solution to fill the pipette. The same caustic potash solution is used as was used in the CO_2 pipette. Air must be excluded immediately after filling the pipette. Some engineers use 10 g of the acid and dissolve it in 20 cu cm of water. This solution is poured into the pipette first, and the caustic solution is then added. The third pipette, for absorbing carbon monoxide, contains a cuprous chloride solution. There are several ways of preparing this solution, but it is rather difficult to prepare a solution that is satisfactory. A common method and one for which the materials are easily obtained, is as follows: Pour about 50 g of black copper oxide (scale) into a 500-cu cm flask and insert also several strips or wires of copper, long enough to reach up to the mouth of the flask. Fill the flask *full* of 1.10 gravity hydrochloric acid (1 part cp acid to 1 part water), and insert stopper. Let stand until the solution is colorless, then pour off the clear liquid. This liquid may be stored in *full* bottles containing copper strips, providing the bottles are well sealed from air and protected from strong light.

EXPERIMENT 76. Boiler-feed-water Analysis

I. PREFACE

The complexity of the problem of feed-water treatment is due largely to the fact that the natural waters available for boiler feeding are varied in their composition, so that the treatment that applies to water from one source is not suitable for waters from another source or locality. The treatment to be applied in any given case must then be determined by chemical analysis. Although the problem of initially prescribing the proper treatment is frequently turned over by the engineer to the industrial chemist, the carrying out of the prescribed treatment is again the engineer's problem, and he must have sufficient scientific knowledge of the subject to control the process and to make any changes demanded by changing conditions. The present-day high operating pressures and the tendency to operate at very high rates of evaporation make it imperative that the engineer keep his boiler-water conditions under strict control.

Although the four difficulties caused by impure feed water (*i.e.*, scale, priming, corrosion, and embrittlement) are usually in themselves apparent upon investigation, it is nevertheless difficult to generalize concerning either the causes of or the remedies for these conditions. Not only do these causes and remedies vary with the water used and with the character of the boiler plant, but infrequently in a given case several causes are operative and several remedies are available.

Scale varies in character from a tenacious flintlike deposit to a soft clay mud. Much of the mud and sludge deposited in a boiler frequently enters as suspended matter in the feed water, and this type of deposit can be largely eliminated by previous filtration of the water. The most common scale-forming materials originally in solution in the water are the carbonates and sulphates of calcium (lime) and magnesium. Carbonate hardness in the feed water produces a relatively soft scale or sludge, whereas sulphate hardness produces a hard, adherent scale. Heat transmission is retarded by scale, with resultant increase in thermal losses and frequently with damage to the scale-covered steel by overheating, especially if the rate of heat absorption is high. Carbonate hardness is usually removed by treating the water with lime. Soda ash (sodium carbonate) is the most common reagent for treating sulphate hardness, though barium salts are sometimes preferred, and caustic soda is also used. Sodium aluminate has recently come into favor for use in connection with lime-soda softeners, to accelerate and complete the softening process. Base-exchange or zeolite softeners will remove both types of hardness, and they have the advantages of compactness, ease of operation, and ready provision of water of zero hardness, but they are not well suited to waters of high carbonate hardness owing to the resulting high concentration of sodium salts in the softened water and to the greater cost of operation over lime-soda softeners for these waters. A combination of lime and zeolite treatment is often advantageous. By carrying an excess of sodium carbonate in the feed water (or by maintaining a certain carbonate-sulphate ratio), it is possible to inhibit the deposition of hard adherent sulphate scale, even though the feed water shows a considerable amount of sulphate hardness. At pressures above 250 lb it is preferable to substitute sodium phosphate for sodium carbonate. The scale problem is of such importance in the high-pressure plant that many of the newer stations distill all make-up water in evaporators provided for this purpose. The evaporator is a necessity in marine practice.

Priming, foaming, and water entrainment in the steam occur in varying degrees from a fraction of 1 per cent to 10 per cent or more and result in wear and clogging of valves, governors, engines, and turbines. Common causes of this difficulty are many, among them being oil, organic matter, suspended material, and a high concentration of soluble salts in the boiler water. Often two or more of these causes operate simultaneously. Once these impurities have been introduced into the boiler, frequent "blowing down" is necessary, to keep the concentration within the boiler below a value which will cause priming. "Continuous blowdown" is resorted to in some cases. The real remedy is of course to keep these substances out of the boiler, as a large amount of blowing down results in an appreciable heat loss.

Corrosion or "pitting" of the boiler steel is most frequent when the feed water contains dissolved oxygen (or air), though it may be caused by acid or other corrosive chemicals in the water. Certain magnesium salts (the chloride in particular), are often responsible for corrosion. Apparently, the usual case of boiler corrosion is an electrochemical process, and, although the exact mechanism of the process is not well understood, certain treatments are known to eliminate it in the majority of cases. A most effective large-scale method, and the one in general use, is to "deaerate" the feed water before it is introduced into the boiler. This is usually accomplished by a special apparatus, but in some cases an open feed-water heater or a surface condenser hot well may function as a deaerator. Corrosion may be reduced to a negligible amount by carrying a certain concentration of sodium hydroxide in the water. Water of low hydrogen-ion concentration is less corrosive than water of high hydrogen-ion concentration.

Embrittlement of boiler plate, rivets, or tubes is a little understood phenomenon, but it frequently occurs in the presence of sodium hydroxide in the feed water. This caustic may not be present as such in the boiler feed, but it may be formed within the boiler itself by the breaking down of sodium carbonate. This theory of embrittlement has become well enough established so that the A.S.M.E. Code covering the Care of Power Boilers makes definite recommendations regarding the permissible amounts of total alkalinity within the boiler at various pressures.

The A.S.M.E. suggested rules for the Care of Power Boilers specify that the following tests shall be made upon the water within the boiler; total solids in solution, alkalinity, chlorides, sulphates, ratio of chlorides to total solids, ratio of chlorides to sulphates, and the ratio of total alkalinity in terms of sodium carbonate to the sulphates in terms of sodium sulphate.

II. BOILER-WATER TESTS

1. Total Hardness by Standard Soap Solution. *a. Instructions.*—To 58.3 cu cm of filtered water add two drops of methyl-orange indicator and titrate with $N/50$ sulphuric acid until the yellow color deepens to an orange tinge. Pour this neutralized water into a test bottle and add standard soap solution slowly, ½ cu cm at a time, shaking after every addition, and finally adding five drops at a time between shakings. The test is completed when the lather will persist for 3 min after shaking. A false end point or "ghost point" is sometimes mistaken for the true end point of the test. In order to check this, a further addition of 5 cu cm of soap solution should be made after the test has apparently been completed, and if the lather then fails to persist for 3 min after shaking, the test should be continued as before, until the true end point is reached. If the total hardness exceeds 10 grains per gallon, the water should be diluted with an equal volume of distilled water, before taking a sample.

b. Results.—The number of cubic centimeters of soap solution used, less 0.5 (the distilled-water factor), equals the total hardness in grains per gallon in terms of calcium carbonate. If a diluted sample is used, the 0.5 is first subtracted and the remainder multiplied by 2 to obtain the hardness.

c. Applications.—When this test is made upon raw water, the quantity of soda ash required for the treatment of permanent hardness may be computed from the results (see A.S.M.E. rules for the Care of Power Boilers, page 83). This is also the most common routine test used in the control of water-softening systems, but in this case the step of neutralizing with acid is often omitted. The following rough classification of waters on the basis of their total hardness and as regards their suitability for boiler feeding is often given: Less than 8 g per gallon, very good; 8 to 15 g, good; 15 to 20 g, fair; 20 to 30 g, bad; over 30 g, very bad. If the water contains much sulphate hardness, these figures are high.

2. Total Alkalinity. *a. Instructions.*—To 58 3 cu cm of filtered water at a temperature not to exceed 100°F, add two drops of methyl-orange indicator and titrate with $N/50$ sulphuric acid until the yellow color deepens to an orange tinge. (The exact point at which the color turns may be recognized by having a duplicate sample adjacent for comparison.)

b. Results.—The number of cubic centimeters of acid used equals the total alkalinity (*i.e.*, carbonate, bicarbonate, and hydroxide) in grains per gallon. If the alkalinity exceeds the total hardness by soap test, the presence of sodium carbonate is indicated, and there is no carbonate hardness.

c. Applications.—When this test is made upon raw water, the quantity of lime required for the treatment of carbonate hardness may be computed from the results (see A.S.M.E. Code, page 83). When this test is made upon a treated water, the effectiveness of the treatment for carbonate hardness is indicated. This is one of the tests used in connection with the maintaining of the prescribed sulphate-carbonate ratios for prevention of caustic embrittlement (see above).

3. Carbonate, Bicarbonate, and Hydroxide Alkalinity. *a. Instructions.*—To 58.3 cu cm of filtered water at a temperature not to exceed 100°F, add two drops of phenolphthalein indicator, which will turn the water pink. Titrate immediately with $N/50$ sulphuric acid, stirring constantly and adding the acid drop by drop until the color has just disappeared. Record the burette reading and mark it P. Add two drops of methyl-orange indicator and, without refilling the burette, continue the titration until the color changes from yellow to orange. Record the total cubic centimeters of acid used for both titrations, and mark it M.

b. Results.—(1) If $P = 0$, the alkalinity is present as bicarbonates only. (2) If P is less than $M/2$, carbonates and bicarbonates are present. (3) If

$P = M/2$, carbonates only are present. (4) If P is greater than $M/2$, carbonates and hydroxides are present. The hydroxides, in grains per gallon, equal $0.8(2P - M)$. The carbonates, in grains per gallon, equal $2.12(M - P)$. (5) If $P = M$, hydroxides only are present.

c. Applications.—When this test is made upon a lime-soda treated water, the presence of hydroxides indicates an excess of lime. Certain natural waters contain both carbonates and hydroxides. The A.S.M.E. rules for the Care of Power Boilers state that the concentrations of free sodium hydroxide and sodium carbonate within all steaming boilers should be kept in excess of 15 g per gallon (in terms of sodium carbonate), in order to avoid certain types of corrosion.

4. Chlorides. *a. Instructions.*—To 58.3 cu cm of the filtered water at a temperature not to exceed 100°F, add two drops of phenolphthalein, and if a pink color appears, titrate with $N/50$ sulphuric acid to a colorless condition. Add a few drops of potassium-chromate indicator (2 per cent solution), which will turn the water yellow; then titrate with a standard solution of silver nitrate (1 cu cm = 1 mg chlorine). The end point has been reached when the first tinge of brownish red or orange appears. This point is rather difficult to detect, and a duplicate sample may be used for comparison, or the sample may be viewed by yellow light (using a yellow curtain or amber glasses).

b. Results.—The number of cubic centimeters of silver nitrate used equals grains per gallon of chlorine.

c. Applications.—When this test is made first on the feed water and then on the blowdown water, the ratio of results will indicate the concentration of the water in the boiler. If, for example, the test on the feed water gives a result of 5 and that on the blowdown water a result of 20, the water in the boiler carries four times the salt concentration found in the feed water.

5. Other Tests. *a. Sulphates by Photometer.*—This test is accomplished by adding barium chloride to the water sample and estimating the sulphates by the amount of barium sulphate precipitated. The photometer consists of a glass graduate with a light mounted behind it. The barium-sulphate solution is poured into the graduate and then diluted until the light is just visible through the turbid mixture. Or a given quantity of water may be first introduced into the graduate and the turbid solution added until the light just disappears. The amount of sulphates is read from a scale on the graduate. The sulphate determination is necessary in connection with the control of sulphate-carbonate ratio, as a precaution against embrittlement.

b. Total Solids and Suspended Solids.—Total solids may be determined by the evaporation of a measured sample in a weighed evaporating dish. The difference between total solids in the unfiltered and filtered samples represents the suspended solids.

c. Oxygen.—In plants using special equipment for the deaeration of the feed water, the oxygen test should be made frequently, but the apparatus and instructions for this test are usually furnished by the manufacturer of the deaerating equipment. The oxygen determination is a chemical test. It is not particularly difficult and may be carried out by an unskilled operator. Some difficulty is encountered in obtaining a representative water sample.

d. Hydrogen-ion Concentration by Colorimeter.—This test furnishes another method of measuring the alkalinity or acidity of the water, though it is not as accurate as the conductivity method. The water sample is treated with an indicator or dye and the resulting color matched against a series of "color standards."

e. Electrical-conductivity Tests.—Alkalinity, acidity, or salt concentration (*i.e.*, the concentration of hydrogen or hydroxyl ions) in a water may be determined from its conductivity. Commercial instruments are now available for conductivity measurements. Perhaps the most important feature of this method of testing is that the instruments may be made recording and thus the condition of the water be under continuous observation. These instruments are widely used on condensate lines, for indicating any inleakage of circulating water from surface condensers (see page 413).

f. Qualitative Test for Calcium.—A filtered sample of water is made alkaline to litmus paper by the addition of ammonia. If a test tube is half filled with this water and five drops of $N/2$ ammonium-oxalate solution is added, the prompt appearance of a fine white precipitate usually indicates high calcium, requiring soda ash for its removal.

g. Approximate Test for Foaming.—A pint bottle is half filled with partly cooled and filtered blowdown water and shaken violently. If a distinct foam persists, there is danger of foaming in the boiler, and the cause for this condition should be sought at once.

EXPERIMENT 77. Test of an Evaporator

I. PREFACE

Evaporators are used in the industries in the manufacture of salt, sugar, condensed milk, and extracts from chemical solutions, in the distillation of water, and in similar processes. In both stationary and marine power plants, the evaporator is used to remove the salts from the boiler-feed waters. Also, certain types of evaporators may serve to secure a proper heat balance for a given power station. Impure boiler-feed water produces scale, corrosion, metal embrittlement, and foaming. Feed waters are often treated chemically to remove certain salts and suspended impurities (see Experiment 76). A high concentration of sodium salts, however, such as found in sea water, cannot be successfully treated by chemical means, and evaporation is employed to remove these salts.

The evaporator to be tested in this experiment is for the distillation of raw make-up water for boiler-feed or domestic use. This is a much simpler problem than is ordinarily met with when weak liquors are evaporated down to heavier concentrations, in which case there are involved varying specific heats, specific gravity, and boiling points in the different effects.

The **single-effect** evaporator is built the same as a closed heater or surface condenser, *i.e.*, it consists of an outer shell containing straight or curved tubes with steam on the one side and water on the other. If the water is made to pass over the tubes in a thin film, it is termed a **film type** evaporator; if the tubes are submerged in the water, the evaporator is termed a **submerged-tube type.** In a third, or **flash type** evaporator, the vapor is flashed into steam upon reduction of its pressure, usually by injection into a vacuum chamber.

Evaporators are also classified according to their initial pressures into high pressure or **high heat level** (using live steam), and low pressure or **low heat level** (using exhaust steam). Further divisions of these classifications are sometimes made.

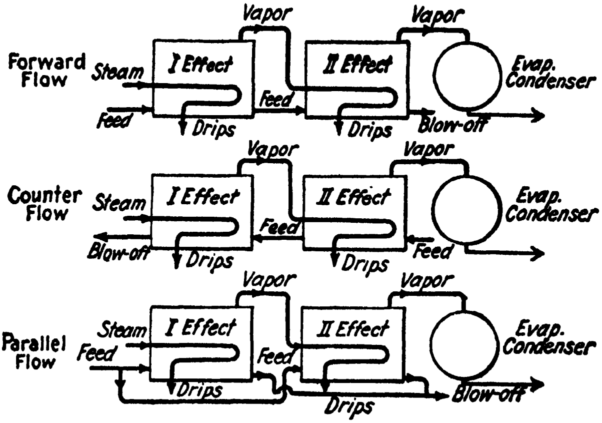

FIG. 185.—Methods of connecting evaporators.

Evaporators may be hooked up for parallel flow (Fig. 185) as in the above case, *i.e.*, raw water is fed independently to the jackets. In series or forward flow (Fig. 185), all the raw water is introduced into the first effect. The second effect receives its jacket water by means of a connection from the jacket of the first effect, then through the trap *R* and to the jacket of the second effect. The pressure difference between the two effects is sufficient to force the water through the trap *R*. In this arrangement, the water fed to the second effect has been previously heated in the first effect.

The evaporator may also be connected for counterflow (Fig. 185), *i.e.*, all the raw water enters the second effect and is then pumped from the second effect to the first effect. Other modifications as to arrangements of heating the incoming raw water, utilizing heat from condenser, and cooling of condensate from the effects may be found in various installations.

The condition for the functioning of an evaporator is that the steam must be at a higher temperature than the water, so that the heat interchange will take place from the steam to the water. The saturated or wet steam will therefore condense, and the water that is subjected to a lower pressure will evaporate. This generated steam, free from any chemical salts (if 100 per cent quality) is led into a condenser and becomes a liquid. This liquid is the finished product, or distilled water; the impurities are left in the evaporator.

When two evaporators are connected in series, the second evaporator utilizes the steam generated in the first effect as the heating medium to vaporize the raw water surrounding its heating coils. This generated steam is led to a condenser, as in the case of a single-effect evaporator. In the **double-effect** evaporator, the second effect serves as a condenser for the generated steam of the first effect. In this case, both effects furnish distilled water. In the **triple-effect** evaporator three evaporators are connected in series, the process taking place in a similar manner as explained for the double-effect type. All three effects furnish distilled water. The number of evaporator effects may be increased to the limit of commercial economy. As many as 12 effects have been used in certain process work. In this case the tail effects are connected to the vacuum of a condenser.

In general, the greater the number of effects, the greater the amount of raw water distilled from any solution per pound of initial or boiler steam. General practice gives the following values for water evaporated per pound of steam: For the single effect, 0.8; double effect, 1.5; triple effect, 2; quadruple effect, 2.5; etc. In stationary plants, in general, not more than three effects are used for the distillation of boiler make-up water.

The sketch of Fig. 186 gives in outline the general arrangement of a simple submerged-tube, double-effect evaporator. Live steam of a known quality is introduced through a reducing valve into the first effect; its pressure P_1 and temperature T_{13} are determined. The water in the jacket of the first evaporator condenses the incoming steam; it leaves as condensate only, which is led through the cooling coils and is weighed as W_1. The steam generated in the first effect at the saturation temperature T_1 passes over through the separator into the second effect as live steam. If the separators are well insulated, the radiation will be small and the weight W_5 almost negligible. From this weight

W_5 in connection with the total steam generated in the first effect, the quality leaving the first effect may be computed on the basis that no condensation takes place in the separators. As a rule, the disengaging area and the volume of the steam space of the effects are large so that the steam quality entering the separators is close to 100 per cent. (It is a very difficult problem to determine the quality of low-pressure steam.) The weight of steam generated in the first effect is that condensed in the second effect and weighed as W_2. The rate of feed of the raw water into the first effect is regulated by the constant-level float trap R. The weight fed in to maintain constant level is the weight W_2, and its entering temperature is T_4.

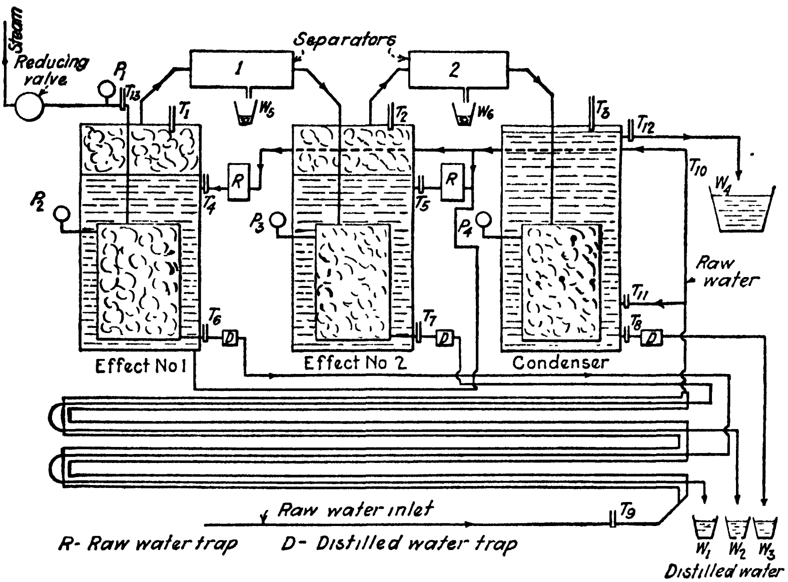

FIG. 186.—Diagram of double-effect evaporator.

The process in the second effect follows identically that of the first effect.

The third container serves as a condenser for the second effect. Circulating water enters the condenser at T_{11} and leaves at T_{12}. The following equation may be readily established, on the assumption that each effect evaporates raw water, or that the effects are connected in parallel.

$$\left\{\begin{matrix}\text{Heat input}\\\text{First effect}\end{matrix}\right\} = \left\{\begin{matrix}\text{heat output}\\\text{first effect}\end{matrix}\right\} + \text{radiation}$$

$$\underset{\substack{\text{Incoming}\\\text{steam, above}\\\text{32 deg}}}{W_1(XL + h)} + \underset{\substack{\text{Incoming}\\\text{water, above}\\\text{32 deg}}}{W_2(T_4 - 32)} = \underset{\substack{\text{Leaving}\\\text{condensate,}\\\text{above 32 deg}}}{W_1(T_6 - 32)} + \underset{\substack{\text{Leaving}\\\text{steam, above}\\\text{32 deg}}}{W_2(X_1L_1 + h_1)} + \underset{\substack{\text{Radi-}\\\text{ation}}}{R.}$$

Similarly, we may build up the equations for the second effect and for the condenser.

II. INSTRUCTIONS

Be sure that the shells or jackets are filled with raw water up to the predetermined level as established by the mark on the jacket gage glass. These levels should be maintained constant throughout the entire test, being automatically controlled by the trap *R*.

Open the live-steam valve, allowing steam to pass through the reducing valve and enter the first effect. Check traps *D* for proper working order by means of their sight gage glasses. The process of distillation is now in operation.

When steam enters the condenser, turn on cooling water, adjusting same later to secure the desired temperatures after constant conditions have been established.

The A.S.M.E. Code calls for manometers for measurement of pressures instead of spring gages.

Form your own blank.

III. DETERMINATIONS

1. The adaptability of the apparatus to its work and the best method of operation.
2. The capacity, gallons distilled per 24 hr.
3. Rate of heat transfer with stated conditions of steam supply, temperature, and pressure.
4. Pounds of steam per pound of distillate or concentrate.

Heating surface = surface in contact with the steam.

Rating based on gallons per hour is unsatisfactory, since this rating may be altered by change in feed, .temperature, steam pressure, vacuum, radiation losses, or other factors. Heat-transfer coefficients are not so much affected by such changes.

$$\text{Apparent heat-transfer coefficient} = U = \frac{H}{A \times t_h(t_3 - t_1)}$$

$$= \text{Btu per sq ft per degree difference per hr}$$

where H = total heat transmitted, Btu in time t_h.
A = heating surface, square feet.
t_h = time, hours.
$t_{1s} - t_1$ = apparent temperature drop, degrees Fahrenheit.

Apparent temperature drop = difference between the saturation temperature of the heating steam (calculated from its pressure), and the temperature of the vapors of the boiling liquid.

$$\text{Thermal efficiency} = \frac{\text{Btu of evaporation}}{\text{Btu of condensation}} = \frac{\text{heat lost by evaporation}}{\text{heat which caused it}}.$$

EXPERIMENT 78. Test of Steam Boiler

I. OBJECT

The object of this test is to determine the efficiency, the evaporation per pound of fuel, and the cost of evaporation for a steam-boiler unit. The heat-balance analysis, or analysis of heat losses, is also to be made.

II. PREFACE

The importance of boiler tests among power-plant tests, as regards their relation to the operation of the plant, has already been emphasized (see Preface to the present division on Steam Generation). A "complete" commercial test of a large boiler involves much detail work, a large test crew, and a period of 8 to 24 hr at each load for the test itself.

When an engineer undertakes to run a complete boiler test he comes to realize that the job involves a considerable amount of responsibility in several respects. The nature of this responsibility depends upon the object of the test, whether it be the checking of compliance with a guarantee, determining the effect of a change in design, comparing different fuels, or merely making an analysis of current plant performance.

The A.S.M.E. Test Code for Stationary Steam-generating Units states that steam-generating unit testing should not be undertaken by anyone who has not had some training under an experienced testing engineer if reliable results are to be expected. The whole subject should be thoroughly understood, both theoretically and practically. Accuracy of the test depends very largely upon the care and faithfulness of the observers. It is much easier to make mistakes than is realized by those who are not familiar with practical testing.

The absolute accuracy of the results of a steam-generating unit test, even when conducted with the greatest care, is doubtful, but there is as yet no possible basis upon which to determine what the probable limits of errors might be. It is generally conceded, however, that there are several sources of indeterminate error, the more important of which are discussed below. The limits of accuracy of a test may very reasonably be taken to be within plus or minus 3 per cent.

This Code gives specific instructions covering all phases of the conduct of a boiler test and of the calculation and reporting of test results. The test called for in this experiment will conform essentially to the requirements of the Code, but certain modifications must of necessity be made on account of the limited time available and the inexperience of the personnel, and because the main purpose of the test is that of instruction. The various observations of the test are closely interrelated, and accurate tests depend to a great extent upon the care and faithfulness of the observers. It frequently happens that an entire 24-hr test must be discarded on account of the inattention of a single observer at some time or other during the test.

It is assumed in the present test that the student is already familiar with the proper methods of handling the test instruments and apparatus, including the Orsat, the steam calorimeter, thermocouple pyrometers and high-temperature thermometers, etc., and that he is acquainted with the accepted methods of coal sampling, ash sampling, and flue-gas sampling. If such is not the case, the various experiments and discussions herein covering these points should be reviewed *before* the student reports to the laboratory. Special attention should be paid to the Precautions given for securing accurate results with the various instruments. In order that the significance of the various observations may be better understood, the student should also make sure that he understands how each item in the tabulation of results is calculated (see Form 78*b*, page 386). A student with neither test experience nor book preparation can hardly expect to be of much more value in a boiler-test crew than any intelligent laborer.

III. PREPARATIONS

The assignment of positions on a boiler test and the preparations for the test on the part of each observer will depend to some extent upon the duration and comprehensiveness of the test, the method of firing, and the local conditions in the plant or laboratory. Since these instructions are intended to cover the shorter elementary or preliminary laboratory tests usually included in junior laboratory courses, as well as the comprehensive tests usually carried out by senior students in mechanical engineering, they will be modified by the instructor as conditions

ASSIGNMENT OF DUTIES*

Position	Duties	Apparatus required
Chief	Start, supervise and stop test. Collect all data, making duplicate copies	Watch
Steam	Observe: steam pressure, calorimeter readings, barometer. Calculate quality	Steam gage, calorimeter and thermometers, steam tables or Mollier chart, data board
Fuel and firing..	Attend to firing, fuel weighing or metering, and boiler-water level Weigh ashes. Collect fuel and ash samples. Observe smoke density	Apparatus for weighing or metering fuel and for weighing ashes. Large tight cans and quart jars for samples. Shovels and tamping iron, data board. Ringelmann smoke chart
Feed water.....	Weigh feed water and observe temperature	Scales and tanks, or calibrated volumetric tanks for water measurement. 220- deg. thermometer, data board
Gas analysis. .	Draw gas samples and make Orsat analyses	Orsat, gas-sampling tank, pump or aspirator, rubber tubing and connectors, data board
Pressures and temperatures.	Observe: (1) boiler-room temperature, dry and wet bulb; (2) draft over fire; (3) draft below damper, (4) flue-gas temperatures; (5) stoker speeds for coal fuel or fuel pressure and temperature for oil or gas fuel	Two inclined draft gages, 220-deg. thermometer, flue-gas pyrometer, data board
Data . ..	Prepare data sheets for chief. Copy and blue-print completed data sheet	Data board and carbon sheet. Drawing board, tracing cloth, and instruments
Graphical log ..	Plot continuous graph of observed data..........................	Drawing board, cross-section paper, and instruments
Sketch.........	Make and blue-print sketches as directed below	Drawing board, tracing cloth, and instruments

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* For further explanation of duties, see next section.

demand. On page 381 is a suggested assignment of duties. Some positions may require two or more men. Each man will be responsible for the apparatus he uses on the first position to which he is assigned, and he is expected to secure and place his apparatus as quickly as possible so that the test may be started.

IV. INSTRUCTIONS

1. Procedure.—A considerable amount of time is often consumed in setting up the test apparatus and in adjusting the load and the fires. The flue-gas analysis progresses rather slowly for an observer not accustomed to the manipulation of the Orsat. These difficulties make it the more imperative that each man on the test squad shall do his utmost to see that *his* job does not hold up the starting or progress of the test. After he has the job well in hand, however, and can spare some time between readings, he should acquaint himself with the work of the other men on the test. It is sometimes impracticable to shift each man from place to place on account of the difficulty of keeping the data straight and on account of the technique required on certain jobs.

The test foreman should get the test under way as soon as possible. He should pay particular attention to the conditions prevailing at the beginning of the test and see that the same conditions prevail at the end of the test. This applies particularly to the fuel bed and fires, the water level, the rate of feeding water and the rate of steaming, but also to the steam pressure, the draft, and the temperature conditions.

These operating conditions are presented in the A.S.M.E. Test Code as follows:

For *hand-fired operation* the furnace should be well heated by a preliminary run at the same combustion rate that will prevail during the test and sufficiently long to heat the setting thoroughly. Burn the fire low and thoroughly clean it, leaving enough live coal spread evenly over the grate (say, from 2 to 4 in.) to serve as a foundation for the new fire. Note quickly the thickness of the coal bed as nearly as it can be estimated or measured, also the water level, the steam pressure, and the time, and record the last as the starting time. Fresh coal should then be fired from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level, the steam pressure, and the time, and record the last as the stopping time. If the water level is lower or higher than at the beginning, a correction should be made by computation, not by feeding additional water. Finally remove the ashes and refuse from the ashpit.

Stokers of the Continuous-dumping Type.—The desired operating conditions, *i.e.*, speed and stroke of coal-feeding mechanism, speed of grate, intensity of draft or blast, and rate of water feed should be maintained as nearly constant as possible for at least 1 hr and preferably for 2 hr, before starting and stopping the test.

Stokers of the Intermittent-dumping Type.—Proceed as above, except that stokers should be cleaned about 1 hr before starting and before stopping the test.

Pulverized, Liquid, or Gaseous Fuels.—To obtain the desired equality of conditions when pulverized, liquid, or gaseous fuel is used, the following method should be employed:

The boiler or boilers to be tested should be operated before the start of the test under the fuel, furnace, and combustion conditions which are to be maintained throughout the test for a period of not less than 3 hr. Fuel, temperature, fuel pressure, and draft conditions should be kept as nearly constant as possible during this period and throughout the test. These conditions for starting and stopping apply for pulverized, liquid, or gaseous fuels only.

In student testing, all these instructions cannot be followed but must be modified to suit the time available for laboratory test period.

Readings are to be taken at 10-min intervals except where the instructor directs otherwise. Readings should always be taken in the same order or sequence, and the order of reading should be indicated on the log sheet. Each observer will prepare his own log sheet before the test is started, keep the same in good form, and sign and hand it to the test foreman at the end of the test.

When coal fuel is used, the coal-handling-and-weighing procedure should be arranged to make possible the determination of coal used at the end of each hour of the test. These hourly figures will serve as a convenient guide in firing and a rough check on the weighing. Coal and ash samples for analysis should be accumulated as the test progresses (see Experiment 72, page 349, for instructions for coal sampling). Care should be taken that the ashpit is clean when the test is started. When oil or gas fuel is used, and the fuel is metered, the temperature and pressure of the fuel at the meters must be included in the data. For methods of smoke determination, see Notes, page 392.

The object of the graphical log is to provide a picture of the test conditions and thus call attention to undesirable or unintentional lack of uniformity, and to bring out errors of observation or record before it is too late to correct them. The data should therefore be collected and the graph plotted immediately after each set of readings. Time should be plotted horizontally and quantities vertically. The graphs of coal and water quantities should be cumulative, but all others will represent instantaneous readings.

The sketches called for should show: (1) the external connections to the boiler, including the piping of the feed-water regulator, the feed pump and pump governor, the feed-water heater and the weighing tanks; (2) a longitudinal section of the boiler and setting showing the boiler construction, location of steam nozzle, feed pipe, water column, and blowdown. This sketch should also show the furnace construction and gas travel and the setting of stoker, grates, or burners and should carry principal dimensions. The over-all dimensions and those of the combustion chamber and boiler drums and tubes should be given.

For additional suggestions on test procedure, see Notes, page 391.

2. Duration of Test.—The A.S.M.E. Test Code recommends that a boiler test should preferably be 24 hr long when coal fuel is used, either hand or stoker fired. Where operating conditions do not permit a 24-hr test, 10 hr is prescribed as the minimum length of test, or, when the combustion is less than 25 lb of coal per square foot of grate surface per hour, the test should be continued until a total of 250 lb per square foot of grate has been burned. For pulverized coal, the minimum is reduced to 6 hr and for oil or gas fuel to 4 hr. Although the present test may not meet the Code requirements as to duration, it should be made as long as the time will permit. The main difficulties in the way of obtaining accuracy in short tests are those involved in securing the same fuel-bed conditions and the same water level and steaming rate at the end as at the beginning of the test. *In a short test, therefore, exceptional care should be exercised in regard to these details.* On account of the heat-storage capacity of the setting, the boiler should be operated under test conditions for some time before the test is started. The Code specifies that this preliminary equalizing period shall be at least 3 hr.

Form 78a.—Steam-boiler Test—Preliminary Data

To determine........................Test conducted by..............
Location...........................Owner.........................
Kind of boiler......................Manufactured by...............
Number of tubes.......Diameter of tubes.........Length of tubes.......
Number of drums......Diameter of drums........Length of drums......
Water-heating surface........Superheating surface.....Grate surface.....
Kind of furnace..............Furnace volume.....Kind of draft.......
Type and maker of stoker or burners ....
Kind of fuel.. ....
Date of test......................Duration of test.................
Fuel data:

Fuel Analysis

Ultimate analysis, percentage	Proximate analysis, percentage	Analysis of gas fuel, percentage
Carbon.................	Fixed carbon.........	....
Hydrogen..............	Volatile matter.	....
Oxygen................	Water..............	....
Sulphur...............	Ash................	....

Baumé gravity, deg. (liquid fuel)....................................
Density at standard conditions of 14.7 lb. and 68 deg., lb. per cu. ft. (gas fuel)..
Heating value at standard conditions (gas fuel) B.t.u. per cu. ft..........
Heating value in B.t.u. per lb. (all fuels)..............................
Cost of fuel per..$.........

Form 78*b*.—Results of Steam-boiler Test

Average pressures, temperatures:

1. Steam pressure, lb. per sq. in. gage.................... ________
2. Barometer, inches mercury... ________
3. Steam pressure, lb. abs....... ________
4. Temperature of steam, deg. fahr..................... ________
5. Quality of steam, per cent.... ________
6. Degrees of superheat......... ________
7. Temperature of feed water, deg. fahr................. ________
8. Temperature of flue gases, deg. fahr..................... ________
9. Temperature of boiler room, dry bulb................. ________
10. Temperature of boiler room, wet bulb................. ________
11. Moisture per lb. air, lb ... .. ________
12. Draft below damper, in. water ________
13. Draft over fire, in. water..... ________

Combustion data:

14. Flue gas analysis, boiler outlet: CO_2___per cent; O_2___per cent; CO——per cent; N_2——per cent
15. Dry gas per lb. fuel, actual.... ________
16. Air supplied per lb. fuel ..... ________
17. Theoretical air per lb. fuel.... ________
18. Percentage excess air......... ________

Hourly quantities:

19. Fuel as fired per hr., lb....... ________
20. Dry fuel per hr., lb......... . ________
21. Water actually evaporated per hr., lb.................. ________
22. Factor of evaporation......... ________
23. Equivalent evaporation per hr.; lb...................... ________
24. Units of evaporation per hr... ________

Horsepower:

25. Boiler horsepower developed.. ________
26. Builder's rating, hp.......... ________
27. Percentage rating developed.. ________

Refuse:

28. Ash and refuse per hr., lb.... ________
29. Refuse, percentage of dry fuel fired .... ________
30. Percentage of carbon in refuse ________
31. Carbon burned per lb. fuel, lb. ________

Evaporation:

32. Actual evaporation per lb. of fuel as fired, lb........... ________
33. Actual evaporation per lb. of dry fuel, lb............... ________
34. Equivalent evaporation per lb. of fuel as fired, lb......... ________
35. Equivalent evaporation per lb. of dry fuel, lb............ ________

Efficiency:

36. Efficiency of boiler furnace and grate, per cent......... ________

Cost of evaporation:

37. Cost of fuel for 1000 lb. actual evaporation,............. $________
38. Cost of fuel for 1000 lb. equivalent evaporation........ $________

Heat balance:

39. Heat to water and steam, per cent.................... ________
40. Heat loss in dry chimney gases, per cent............ ________
41. Heat loss due to moisture in fuel, per cent.............. ________
42. Heat loss due to water from combustion of hydrogen, per cent................. ________
43. Heat loss due to moisture in air, per cent............. ________
44. Heat loss due to incomplete combustion of carbon, per cent.................... ________
45. Heat loss due to unconsumed combustible in refuse, per cent.................... ________
46. Heat loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted for, per cent............. ________

Date........ .... Observers:.... ... ...

3. Calculation of Results.—The tabulation of results, for which a form is given herewith (Form 78), is to show the averages, totals, and calculated quantities for the entire test. Most of the items have already been discussed in the class work in heat engines and thermodynamics. Certain definitions and formulas are repeated here for convenience.

Heating surface shall consist of that portion of the surface exposed to both the gases being cooled and the fluid being heated at the same time, computed on the gas side.

Furnace volume is the cubic space provided for the combustion of fuel before the products of combustion reach any heating surface.

Equivalent evaporation is the amount of water that would be evaporated from water at 212° to steam at 212° by the same amount of heat as was actually absorbed by the water and steam under operating conditions.

Factor of evaporation is the ratio of the heat actually absorbed by 1 lb of water and steam to 970.4. It is therefore the factor by which actual evaporation is multiplied to obtain "equivalent evaporation."

A **boiler horsepower** corresponds to an equivalent evaporation of 34.5 lb of steam per hour.

Rated horsepower is usually determined on the basis of 10 sq ft of heating surface for water-tube boilers and 12 sq ft of heating surface for fire-tube boilers.

The method of calculating the weight of *dry* flue gases per pound of fuel was covered in Experiment 75, Flue-gas Analysis, page 363. This method involves the conversion of the volumetric analysis to gravimetric, but this conversion may be included in the final calculation if desired, *viz.:*

$$\frac{\text{Pounds dry flue gas}}{\text{per pound fuel}} = \frac{\text{pounds carbon}}{\text{per pound fuel}} \div \frac{\text{pounds carbon}}{\text{per pound flue gas}}$$
$$= \frac{\text{pounds carbon}}{\text{per pound fuel}} \times \frac{44CO_2 + 32O_2 + 28(CO + N_2)}{12(CO_2 + CO)}$$
$$= \text{item (31)} \times \frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}. \qquad (68)$$

In the above formula the volumetric flue-gas analysis is to be substituted for the symbols CO_2, O_2, CO, and N_2.

The dry *air* supplied per pound of fuel is equal to the dry gas per pound of fuel *less* the carbon burned *plus* the oxygen which united with the hydrogen in the fuel and thus disappeared in the condensed water vapor from the flue gas. In equation form:

$$\frac{\text{Pounds air}}{\text{per pound fuel}} = \frac{\text{pounds dry flue gas}}{\text{per pound fuel}} - \frac{\text{pounds carbon}}{\text{per pound fuel}} + \frac{8(\text{pounds hydrogen})}{\text{per pound fuel}}. \qquad (69)$$

It is to be noted that the "pounds of carbon per pound of fuel" referred to in the above formulas is the carbon *burned* per pound of fuel (item 31, page 386), or the carbon by analysis less the carbon in the ash and refuse.

Another method of calculating the air supplied per pound of fuel is based on the nitrogen content of the flue gas and has the advantage of not requiring the hydrogen content of the fuel by analysis The following ratios are used in this method: (1) pounds of nitrogen per pound of flue gas; (2) pounds of nitrogen per pound of air; (3) pounds of flue gas per pound of fuel. These ratios may be arranged as follows:

$$\frac{\text{Pounds nitrogen}}{\text{per pound flue gas}} \times \frac{\text{pounds air}}{\text{per pound nitrogen}} \times \frac{\text{pounds flue gas}}{\text{per pound fuel}} = \frac{\text{pounds air}}{\text{per pound fuel}}. \qquad (70)$$

It is noted that a cancellation on the left side of the equation leaves "pounds of air per pound of fuel." A number of other methods are available for calculating the air supplied.

The air theoretically required per pound of fuel is calculated from the combining weights:

$$\text{Theoretical air} = \frac{32}{12} \times \frac{1}{0.23}\text{C} + \frac{16}{2} \times \frac{1}{0.23}\left(\text{H} - \frac{\text{O}}{8}\right) + \frac{32}{32} \times \frac{1}{0.23}\text{S}$$

$$= 11.6\text{C} + 34.8\left(\text{H} - \frac{\text{O}}{8}\right) + 4.35\text{S}. \qquad (71)$$

The symbols C, H, O, S, and N are here used to represent the carbon, hydrogen, oxygen, sulphur, and nitrogen content of the fuel as shown by the ultimate analysis.

The **water actually evaporated** is the water fed to the boiler less the water in the steam (corrected for leaks and blowdown, if any).

A **unit of evaporation** according to the Code is 1000 Btu absorbed by the steam per hour, and item 24 (page 386) is therefore "thousands of Btu absorbed per hour."

It is to be noted that, in accordance with the A.S.M.E. Code, all calculations involving the fuel are carried out on the basis of 1 lb of fuel, whether the fuel be solid, liquid, or gas.

The **heat balance** is most conveniently calculated on the basis of 1 lb of fuel and then converted to the percentage basis. The first five losses are all flue-gas losses, but they are computed separately.

In computing the **heat lost in the dry flue gases,** the weight of dry flue gas per pound of fuel is used rather than the weight of air, and the loss is that heat required to heat this weight of gas from boiler-room temperature to stack temperature, using a mean specific heat of 0.24.

All three **losses due to moisture** in the flue gas are computed in a similar manner. The specific heat of superheated steam in this range is taken as 0.46. The loss due to moisture in the air is heat of superheat only. The loss due to moisture in the fuel includes the heat of the liquid from the fuel temperature to the vaporizing temperature, and the latent heat at the partial pressure corresponding to the vaporizing temperature as well as the heat of superheat from the vaporizing temperature to the stack temperature.[1] In simplified form the equation reduces to

$$h = w(1{,}089 + 0.46T_g - T_f), \tag{72}$$

where w is the weight of moisture in 1 lb of fuel, T_g is the temperature of the flue gas, and T_f is the temperature of the fuel. Similarly, the hydrogen loss equals

$$h = 9\mathrm{H}(1{,}089 + 0.46T_g - T_f), \tag{73}$$

where H is the weight of hydrogen in 1 lb of the fuel.

The hydrogen loss in the case of natural gas is higher than when burning oil or coal.

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[1] For further explanation, see Gebhardt, "Steam Power Plant Engineering," p. 94.

The **loss due to incomplete combustion** of carbon is negligible in a well-designed and properly operated furnace. This loss is computed by the formula

$$h = \frac{CO}{CO + CO_2} \times C \times 10{,}160, \qquad (74)$$

where CO and CO_2 represent volumetric percentages of these flue-gas constituents and C is the pounds of carbon *burned* per pound of fuel; 10,160 is the heating value of 1 lb of carbon when contained in carbon monoxide. The derivation of this equation is evident from the fact that the same weight of carbon is present in a molecule of CO as is present in a molecule of CO_2 and hence the fraction in Eq. (74) represents the ratio of carbon in the CO to total carbon in the flue gas.

The final item of the heat balance is obtained by subtracting the sum of all the others from 100 per cent and serves as a rough check on the accuracy of the test work.

The heat balance as discussed above is what the Code calls the "short form." It shows the total or gross amount of each of the boiler losses. A further refinement is to separate from these losses the "unavoidable" losses (see A.S.M.E. Test Code, Table *S-2-b*). This distinction is a vital one to the operating engineer, as he is primarily concerned with reducing the "avoidable" or preventable losses. The importance of this distinction may be realized from the fact that in the average plant the heat loss in the dry stack gases is usually over 20 per cent; whereas, in central-station operation, where avoidable losses are studied, it usually does not exceed 10 per cent.

4. Report.—As indicated above, the original observed data for this test are to appear in the report, both as a tabulation and in the form of a graphical log.

The discussion is to include a consideration of the degree of accuracy of the test, the sources of error and the relative weight of each, and suggested modifications for securing greater accuracy. The student should give his opinion as to the quality of the performance shown by the test, both with respect to over-all efficiency and with respect to the magnitude of the individual losses.[1] The student should also suggest any operating improvements or repairs which in his opinion would improve the efficiency of the boiler room. These are to be listed in the order of their importance, and the estimated improvement to be effected by each should be given. If additional instruments or more complete operating records are desirable, definite recommendations should be made. It must be kept in mind, however, that boiler-room instruments, with recording and integrating features, range in price from $50 to $500 or even more, and that an engineer must be fully justified before recommending such expenditures. Many factory executives and building managers do not look with favor upon these expensive instruments, because they have seen elaborate installations depreciate into worthless junk through the neglect of the operators. Such instruments require a certain amount of expert maintenance. In large plants this work is usually taken care of by one or more of the younger technical men, but in many small plants there are no technical men employed. If the instruments are neglected, their indications become inaccurate, and in many ways inaccurate instruments are worse than none at all. Such factors as these should be taken into account when making recommendations.

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[1] See pp. 344–346 and 393–396.

V. NOTES

In order to include certain additional material on test methods and to show more clearly the significance of the test results, the following notes have been prepared.

1. Notes on Test Methods. *a. Coal.*—It is recognized that one of the greatest possibilities of error in the entire test is that in connection with the collecting of a fuel sample and the reducing of this sample to a quantity which can be handled in the laboratory. The methods already outlined should be carefully followed.

b. Water.—The weight of water evaporated is determined by weighing or measuring the volume of the water supplied to the boiler during the test, taking care that the gage-glass water level and the steaming rate are the same at the end as at the beginning of the test. For volumetric measurements, only special narrow-neck tanks should be used (see Fig. 35, page 57). These should be carefully calibrated before the test, and in using them the water temperature should always be observed. Water measurement by meter is not recognized as sufficiently accurate for boiler tests. There are times, however, especially in large installations, where metering must be resorted to. In such cases either the venturi meter or the V-notch weir meter is to be preferred, and the precautions given in Chap. VII, page 301, should be carefully observed. These meters should be used in the boiler-feed line and not in the steam line, as steam meters are difficult to calibrate and their accuracy is greatly affected by variations in the pressure, temperature, and quality of the steam. A steam meter has one important use in a boiler test, and that is to furnish a guide to aid the operator in maintaining a uniform rate of evaporation.

c. Flue Gas.—Assuming that the technique of handling the Orsat apparatus has been mastered, the chief difficulty in accurately determining the composition of the flue gases is that of obtaining a representative sample. Various forms of sampling tubes have been advocated, including radial "spiders" with open-end spokes, perforated pipes, manifolds with several open-end pipes, and single open-end pipes. The single pipe, with its open end placed in the center of the gas stream, is usually favored. To keep sampling lines open, they should be blown out at intervals with steam or compressed air. In drawing the gas from the flue it is preferable to use a steam, water, or air aspirator, operating continuously during the test period. The sampling lines are thereby always kept filled with "fresh" gas. A collecting tank is sometimes used for storing several samples taken at intervals or for collecting a continuous sample over a considerable period of time. A single analysis will then represent the approximate average over the total time period. When a continuous record is wanted, some form of continuous CO_2 machine is preferred, and this is in turn checked by Orsat analysis of "grab" samples.

d. Temperatures.—Gas temperatures at the boiler outlet may be made with mercury thermometers, gas- or vapor-filled expansion thermometers, or thermocouple pyrometers (see Experiment 4, page 33). The instrument should be located in what the observer considers to be the center of the gas stream. The difficulties encountered in this measurement are chiefly those due to radiation effects and to the fact that the gas travels at different velocities in different parts of the stream and is in addition more or less stratified. These difficulties often make it necessary to do some experimenting before satisfactory locations for the thermometer bulbs are found. If a sectional drawing of the boiler setting is examined, the flow of the main gas stream can usually be determined and the thermometers located accordingly.

e. Smoke.—If arrangements can be made so that the boiler under test is served by a separate stack, smoke observations should be made during the test. There is no universally accepted method of smoke measurement. The method used by many cities in connection with smoke abatement is to compare the appearance of the smoke with the appearance of the various areas on a "Ringelmann Smoke Chart." The areas on this chart are actually ruled into small squares, the size of squares in each area being the same, but the thickness of line varying from 1 to 5.5 mm. The chart is placed at a distance of 50 ft from the observer and as nearly in line with the chimney as possible. At this distance the chart areas appear to be different shades of gray, ranging from a light gray to almost black, and these shades are compared with the smoke issuing from the chimney. Observations are made for 15-sec intervals and are recorded as "percentage of smoke density," each number or unit on the charts having a value of 20 per cent. That is, a smoke corresponding to No. 2 chart is recorded as having a density of

40 per cent, or that corresponding to a No. 3 chart as having a density of 60 per cent. The half-numbers are estimated; in other words, the percentage of smoke density is always expressed in even tens—10 per cent, 20 per cent, 30 per cent, etc. The Code suggests an arbitrary scale of percentages

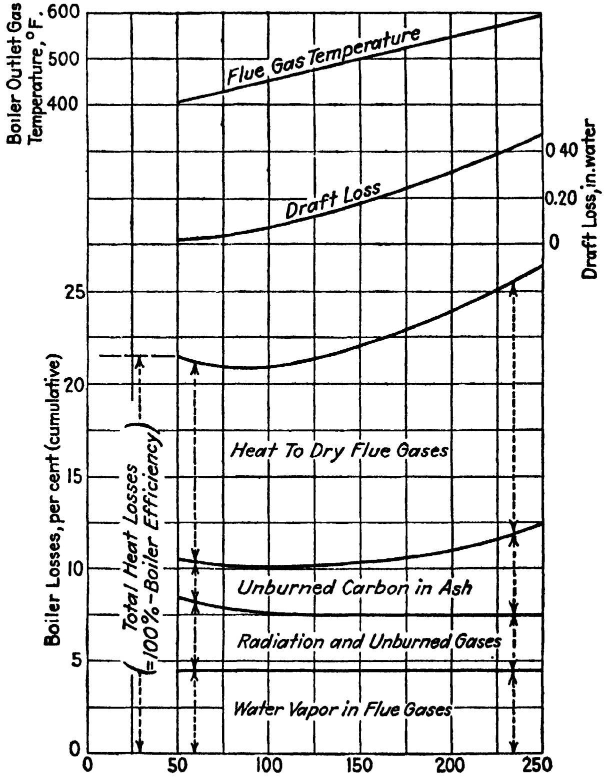

FIG. 187.—Curves showing variation of boiler performance with load.

based entirely on the observer's characterization of smoke color. The following table is given:

TABLE XXIX.—SMOKE PERCENTAGES

Dense black	100	Light gray	20
Medium black	80	Very light	5
Dense gray	60	Trace	1
Medium gray	40	Clear chimney	0

Although this method is necessary in some cases, such as that of a moving locomotive, and it may be quite satisfactory for a trained observer, or one who has had a great deal to do with boiler plants, it is hardly applicable in student work. Personal judgment and the character of the sky, wind, and weather greatly affect smoke observations, even when a standard chart is used, but smoke observations of any type are of some value, especially when plotted in graphical form.

2. Notes on the Reduction of Boiler Losses.—Table XXX and Fig. 187 will serve as a guide in the interpretation of efficiency results and the allocation of losses.

a. Losses Indicated by Low CO_2.—The theoretical CO_2 percentage for complete combustion of coal fuel varies from almost 21 per cent for anthracite to about 18 per cent for certain bituminous coals. The corresponding theoretical percentage is about 15 per cent for oil fuel and about 12 per cent

TABLE XXX.—TYPICAL BOILER EFFICIENCIES

Test efficiencies under usual operating load; good average conditions

Type of service	Heating surface, sq. ft.	Fuel and firing	CO_2 in flue gas, per cent	Flue-gas temperature, deg. fahr.	Heat to water and steam, per cent
Central station, with economizer or air heater	14,000	Stoker	13.5	375	84
	14,000	Pulverized coal	14	375	85
Isolated station, large industrial, institutional, etc.	7,500	Stoker	12	500	77
	7,500	Pulverized coal	13	500	80
	7,500	Oil	11	500	80
	7,500	Natural gas	9	500	80
Small power or heating plant	2,000	Stoker	11	600	73
	2,000	Hand fired	9	600	63
	2,000	Oil	10	600	75
	2,000	Natural gas	8	600	75
Heating; large domestic plant	500	Stoker	9	500	65
	500	Hand fired	8	500	60
	500	Oil	9	500	70
Heating; small domestic plant	100	Stoker	7	450	63
	100	Hand fired	6	450	55
	100	Oil, special design	7.5	350	75
	100	Oil, conversion burner	6	450	65
	100	Natural gas, special design	6	300	75
	100	Natural gas, conversion burner	4.5	450	65

for natural gas. With coal fuel it is seldom advisable to operate with more than 75 per cent of the theoretical CO_2 percentage, on account of the difficulty of eliminating CO when higher percentages are carried. This difficulty is less pronounced with gas, oil, or even powdered coal on account of better mixing. A low CO_2 reading shows a large amount of excess air diluting the products of combustion and carrying away heat. Some of the common causes of low CO_2 may be listed as follows:

1. Leaks in the setting, either through brickwork cracks or around inspection doors, firing doors, etc.

2. Greater draft and larger damper openings than are necessary to furnish the proper amount of air.

3 Holes in the fire (*i.e.*, in the fuel bed on the grate).

4. Short-circuiting of air from the ashpit either around the grates or through that part of the grate area carrying the ashes.

5. Improper burner adjustment or poor burner design or installation when burning oil or gas fuel. (This condition exists especially in the many homemade burner installations.)

6. Stratification and insufficient mixing of air in the combustion chamber, due to poor furnace design and necessitating a large excess of air to eliminate CO.

Few boilers are operated with so low an air excess that CO is found in the flue gases. It is sometimes present when too thick a fuel bed is being carried or immediately after firing in case of a hand-fired setting using highly volatile coal. A small percentage of CO in the gases indicates a large fuel loss, however, and, if a CO content of over 0.2 or 0.3 per cent is found, steps should be taken to correct the condition.

In connection with this discussion of the losses indicated by the gas analysis, it might be mentioned that one of the great difficulties in utilizing instrument readings and test results as a guide to boiler operation is caused by the time required to obtain the figures. The records or instrument indications are not available until some time after the performance itself, and their value for control purposes may then be small because new conditions have arisen in the meantime, such as a different load or a slight difference in the grade of fuel. The operator is in need of methods of measurement that will show him what he is doing **at the time he is doing it.** This difficulty is perhaps more pronounced in the case of CO_2 instruments than of other instruments, but there are now a few CO_2 machines on the market which have a negligible time lag. The "steam-flow air-flow meter" (Bailey) is very satisfactory in this respect.

b. Losses Indicated by High Flue-gas Temperature.—Boiler-stack-gas temperatures are usually within the range of 300 to 600°F. Low gas temperatures accompany light-load conditions and installations in which the heating surface is large. When large economizers or air preheaters are used, the stack-gas temperature may even go below 300°, while with high rate of driving and a small ratio of heating surface to grade surface 600° may be exceeded. It is of course desirable to keep the stack temperature as low as possible, provided this is not done by diluting the combustion products with a large excess of air (a condition that will be shown at once by low CO_2). Among the causes for high stack-gas temperatures may be mentioned:

1. Leaky baffles, allowing part of the gas stream to short-circuit across one or more passes.
2. Insufficient or poorly distributed heating surface.
3. Improper location of baffles.
4. High steam pressures and temperatures.
5. Scale on the water side or soot on the gas side of the heating surfaces, or both.
6. Excessive forcing of fires, *i.e.*, operating at high boiler ratings.
7. Combustion of CO to CO_2 in the boiler passes, known as "secondary combustion."

c. Losses Due to Carbon in the Ash and Refuse.—The ashpit losses accompanying the use of solid fuels are due largely to unburned carbon going to the ashpit. In fact, the loss due to sensible heat in the hot ashes is usually neglected, and the ashpit loss determined entirely by making an analysis of the ash and refuse. When a laboratory sample of the dry ashpit refuse is ignited in a crucible, the loss of weight is recorded as carbon and the heating value calculated accordingly, carbon having a heating value of 14,600 Btu per lb. Ashpit losses usually range from 1 to 5 per cent. The unconsumed carbon in the ash by analysis will range between 10 and 50 per cent. Some of the causes of high ashpit losses are:

1. High ash content in the coal, by analysis.
2. Excessive forcing of fires.
3. Formation of clinker. (Clinker occurs when the ash content of the coal has a low fusing point but may be largely avoided by proper furnace construction and proper firing.)
4. Insufficient time allowed for travel of coal from coking zone to ash zone, resulting in dumping of incandescent carbon into the ashpit.
5. Carrying of "thick spots" in the fuel bed.

d. "Radiation, Unburned Hydrocarbons and Unaccounted For."—These losses are usually reported under one heading, and their combined value is obtained by difference when a boiler heat balance is being calculated. If erratic values are found for successive runs, the accuracy of the test work is subject to criticism. Numerical values for this quantity may be within a range of about 2 to 10 per cent, depending largely on the size of the unit and the extent to which the setting is insulated. It is larger for a small unit than for a large one. Close approximations of the heat loss from the setting by radiation may be made by the methods discussed in Chap. V, page 204. This heat loss does not vary greatly with the boiler load so that the percentage loss will be somewhat higher at light loads.

VI. PRECAUTIONS

a. Use the high heating value of coal in all computations.

b. Since plotted data can be more readily visualized than when in the form of tables it is preferable to have the data from each particular station plotted as curves so as to check the trends from normal. These curves serve as guides in determining errors introduced by an improper setup or discrepancies due to the instrument or personal errors of the observer.

c. Do not follow a high rating test by a low rating test. The soaked-up heat from the setting will continue to release its excess heat at the lower rating until stable equilibrium has been reached by the furnace. Likewise sufficient time must be allowed for each change in rating to permit the furnace to establish stable conditions before test readings are taken.

PART II. STEAM UTILIZATION. TESTS OF STEAM ENGINES, TURBINES, AND CONDENSING EQUIPMENT

EXPERIMENT 79*A*. Mechanical-efficiency Test—Simple Steam Engine

I. OBJECT

The important results to be obtained in this test are the indicated horsepower, the brake horsepower, the friction horsepower, and the mechanical efficiency of a simple steam engine. A further object is to call attention to some of the difficulties involved in the accurate determination of indicated and brake horsepower, so that these may be anticipated when the complete engine-economy and efficiency test of Experiment 79*B* is undertaken.

II. PREPARATIONS

1. Apparatus.

Pressure gages.
Stroke counter.
Steam-engine indicator.
Planimeter.
Platform scales and weights.
Steam engine with friction brake.

2. Setting Up and Starting.—(1) Attach throttle and exhaust-pressure gages, and stroke counter. (2) Attach brake, and set the same on platform scales. Determine tare or dead weight of brake, also length of brake arm. (3) Attach indicator, taking care that the drum does not strike the stops at either end of the stroke. (4) See that all lubricators are filled and in order. Open exhaust valves to atmosphere. Open drain valves, then crack throttle valve to warm up the engine. (5) When all the above are completed, start the engine slowly, and, when it is up to speed, close drain valves.

III. INSTRUCTIONS

1. Determinations.—Calculate load on brake scales for rated load on engine, then consult instructor regarding test runs to be made. A single test of at least 1 hr duration, at rated load, may be desirable, or fractional-load runs may be made. Take all readings every 5 min.

2. Results.—Tabulate data and results under the following heads, giving values for each 5-min period as well as averages and totals for each run: steam pressure at throttle, exhaust pressure, revolutions per minute, gross load on scales, net or live load on scales, brake horsepower, indicated horsepower for head end and for crank end, total indicated horsepower, friction horsepower, mechanical efficiency. Give engine rating, instrument numbers, and miscellaneous data. Show a sample of each calculation, and include a specimen indicator card properly marked and mounted. Draw performance curves of power and efficiency against load if more than one load is used.

EXPERIMENT 79*B*. Economy and Efficiency Test—Simple Steam Engine

I. PREFACE

Much industrial power is generated by steam engines, for neither the steam turbine nor the oil engine can compete in economy with the steam engine in small installations in which there is a simultaneous demand for power and for heating steam.

Steam engines operate under a wide variety of conditions, but the present test will assume that the engine operates noncondensing and that its power may be measured by means of a mechanical brake. When a direct-connected engine generator is to be tested, the test should be modified to conform to the methods used in Experiment 80, Turbogenerator Test, page 403.

II. PREPARATIONS

A suggested outline of duties and a list of the apparatus required by each observer is shown in the table on page 399. Modifications will be dictated by the conditions of the test. If additional men are available, they may well be used as calculators.

The chief should draw up a complete data sheet or assign someone to do this for him. He will start and stop the runs, call time for each reading, and between readings collect all data as follows. From position 3: throttle and exhaust pressures, throttle or calorimeter temperatures, degrees superheat or per cent quality, total revolutions and revolutions per minute. From position 4: gross and net weight on brake scales. From position 5: net weight of condensate. From positions 6 and 7: mean effective pressures, indicated, brake, and friction horsepowers. This general data or log sheet should be complete as soon as the test is finished, including the averages and totals for each run, and a copy of this sheet is required in the report.

Position number	Duties	Apparatus
1	Chief, starter, data collector...	Watch, data board, and carbon sheets
2	Indicator cards...............	Indicator and cards
3	Pressures, temperatures, and speed.......................	150-lb. pressure gage, 15-lb. pressure gage, throttle thermometer or calorimeter, stroke counter
4	Brake horsepower, lubrication..	Friction brake, scales and weights, bearing and cylinder oils
5	Weight of condensate.........	Weighing tanks and scales
6	Planimeter and calculation....	Planimeter, indicator scales, slide rule
7	Planimeter and calculation....	Planimeter, indicator scales, slide rule
8	Sketches, graphical log, data, and result sheet............	Drawing board, instruments, and tracing cloth

Each observer will keep his own data sheet and between readings will calculate any values called for in the previous paragraph.

The observer in position 3 should specify double-spring, heavy-duty gages and a reciprocating continuous counter. A rotating revolution counter may be used if necessary, but a tachometer should not be used without calibration.

The instructions for starting up as given in Experiment 78, page 397, should be followed.

The instructor will specify what sketches and tracings are to be prepared by the man taking position 8.

III. INSTRUCTIONS

1. Determinations.—Make at least four runs, including those at one-half, three-quarters, and full rated load. The A.S.M.E. Test Code specifies that each run should be at least 1 hr long. If the time available for this experiment necessitates shorter runs, all readings should be taken every 5 min. With the brake removed from the engine, two or more "friction cards" should be taken.

2. Results and Calculations.—Form 79 gives a suggested tabulation of results. The averages or totals for each complete run should be shown on these sheets.

The efficiency and economy results of a steam-engine test may be expressed on the basis of either the indicated horsepower or the brake horsepower. In complete tests both sets of results are calculated, if it is practicable to measure both the indicated and the brake horsepower.

Thermal efficiency of the actual engine is defined as follows:

$$\text{T.E.} = \frac{\text{indicated work in Btu}}{\text{heat supplied to cylinder}} = \frac{2{,}545}{S(H_1 - h_2)}, \qquad (75)$$

where S = steam rate, pounds of wet (or superheated) steam per horsepower-hour.

H_1 = enthalpy of steam at the throttle.

h_2 = enthalpy of the liquid at the saturation temperature corresponding to the exhaust pressure.

Rankine cycle efficiency is based on the theoretical Rankine cycle:

$$\text{R.C.E.} = \frac{\text{available energy}}{\text{heat supplied}} = \frac{H_1 - H_{2R}}{H_1 - h_2}, \qquad (76)$$

H_{2R} is the enthalpy of 1 lb of steam which has been expanded along an isentropic line (constant entropy adiabatic), from the actual initial steam conditions to the actual exhaust pressure. This expansion can be followed on a Mollier diagram, or H_{2R} may be calculated from the tables if a Mollier diagram is not available.

Rankine cycle efficiency ratio or "engine efficiency referred to the Rankine cycle," as it is called in the test Code, is defined

Form 79.—Results of Steam-engine Test

Engine built by.......................... Type.......................
Diameter of cylinder..............in. Length of stroke...............in.
Diameter of piston rod............in. Length of brake arm..........ft.
Kind of indicator used.................... Barometer............in. Hg

Item number	Item	Run No. 1	2	3	4
1	Duration of run, min......................				
2	Revolutions per min......................				
3	Steam pressure at throttle, ga., lb. per sq. in..				
4	Steam quality at throttle, per cent..........				
5	Exhaust pressure, ga., lb. per sq. in..........				
6	Total condensate, lb......................				
7	Total dry steam, lb......................				
8	Dry steam per rev., lb....................				
9	Net load on brake, lb....................				
10	B. hp.,...............................				
11	Cut-off, per cent stroke, head end...........				
12	Cut-off, per cent stroke, crank end...........				
13	M.e.p., head end, lb. per sq. in.............				
14	M.e.p., crank end, lb. per sq. in.............				
15	I. hp., head end.......				
16	I. hp., crank end.........................				
17	I. hp. total...........................				
18	Friction hp...........................				
19	Dry steam per b. hp-hr., lb................				
20	Dry steam per i. hp-hr., lb................				
21	Mechanical efficiency, per cent..............				
22	Thermal efficiency (i. hp.), per cent.........				
23	Efficiency of Rankine cycle, per cent.........				
24	Rankine cycle efficiency ratio (i. hp.), per cent				

Date................ Observers:.

as the ratio of the actual thermal efficiency to the Rankine cycle efficiency. It will be noted that this ratio reduces to

$$\text{R.C.E.R.} = \frac{2{,}545}{S(H_1 - H_{2R})}. \qquad (77)$$

3. Report.—The following curves should be plotted against horsepower on the abscissas: (1) steam rate per brake-horsepower hour; (2) steam rate per indicated-horsepower hour; (3) Willans line based on indicated horsepower, *i.e.*, total steam per hour against indicated horsepower developed. The Willans line will serve to call attention to any erratic test results, as the total steam should plot a straight line for a throttling engine or a smooth curve slightly concave upward for an automatic engine; (4) thermal efficiency based on indicated horsepower; (5) Rankine cycle efficiency ratio based on indicated horsepower.

The discussion and conclusions should cover such points as accuracy of tests, reasons for inaccuracies, and suggestions for improvement. If guarantee figures or previous test results are available, the results should be compared with them and the performance of this engine should also be compared with the results for similar machines given in Table XXXII, page 410.

IV. NOTES

1. Scope of Test.—The above test instructions are not intended to cover a complete Code test of a steam engine, such as would be run for an acceptance test under a contract guarantee. It is assumed that this is one of the first efficiency and economy tests run by the student, and several details and precautions incident to a complete formal test are therefore omitted. A very similar and more complete test is the Turbogenerator Test of Experiment 80, following. Several of the determinations included in a complete engine test, such as the determination of engine clearance and leakage, and the full analysis of the indicator diagram, have already been covered in Chaps. II and III, pages 86, 131. For further instructions on steam-engine testing, see the A.S.M.E. Test Code.

2. Measurement of Steam Used.—In the above discussion, the use of a surface condenser and weighing tanks for steam measurement has been assumed. Condenser leakage may affect the accuracy of this method (see page 411), but it is usually preferred as being the most accurate method available. At end of test the water level in the condenser hot well should be at the starting level. Other ways of measuring the steam used are as follows:

a. Volumetric Measurement of Condensate.—This method differs from the one used in this experiment only in the way in which the condensate is measured. If calibrated, narrow-neck, volumetric tanks are used (Fig. 35, page 57) and the water temperature is observed, this method is as accurate as weighing.

b. Weighing the Feed Water Entering the Boiler.—This is of course applicable only when the engine and boilers can be isolated, and when the steam consumption of the auxiliary units served by the boilers can be accurately determined. Careful measurements should be made of water wasted by separators and drips or by leaks of any kind between the point of measurement and the engine throttle. The boiler-water level and rate of steaming must be kept constant. The test Code states that, "with suitable precautions," this method, involving either the weighing or the volumetric measuring of feed water, should give results accurate to within plus or minus 3 per cent.

c. Flowmeter Measurements.—The use of flowmeters is not recognized by the Code, but sometimes it must be resorted to. If calibrated venturi or V-notch weir meters are used, and exhaust-steam condensate is measured, satisfactory results may be obtained. Steam meters should not be resorted to, as they will not register properly under the pulsating flow produced by the engine in the steam-feed line (see Experiment 61, page 301).

d. Steam Consumption Calculated from the Indicator Card.—Under certain conditions this method does not even approximate the actual steam consumption of an engine and it is likely to be misleading unless its limitations are thoroughly understood.

EXPERIMENT 80. Test of Turbogenerator

I. PREFACE

In practically all central stations and other large power plants the prime movers are steam turbines, and the turbines drive electric generators. Although the use of turbines for driving other machines (such as centrifugal pumps, fans, and compressors) should not be overlooked, performance tests on these machines are rather infrequent, and usually the installations are relatively small. In the central station, the use of the bleeder turbine, with stage heating of feed water by steam extracted from the main units, has greatly lessened the importance of the steam turbine as a drive for auxiliaries such as pumps and fans.

The present experiment is therefore concerned with the typical case of a steam-turbine performance test when the load on the turbine is an electric generator. Such a turbine usually operates condensing, and, although the present discussion assumes condensing operation, it will apply equally well if the exhaust pressure is atmospheric, provided the means are available for obtaining the weight of steam used by the turbine.

There are four common methods or terms for expressing the performance of a steam turbine:

1. The **steam rate** or pounds of steam per kilawatt-hour or per horsepower-hour.

This is the prevailing way of expressing performance. There are several reasons why this value is insufficient to show the performance of the machine. The steam rate is affected by changes in inlet steam conditions or in exhaust pressure, and steam rates are therefore not comparable for unlike steam conditions. For a bleeder turbine, the steam rate depends on the amounts of steam extracted.

2. The **heat rate** or Btu per kilowatt-hour or per horsepower-hour.

This rate is a much more logical performance measure. It has, however, the disadvantage of being more difficult to calculate, and it is not always readily understood by the practical operator.

3. The **thermal efficiency** or ratio of heat output to heat input.

This method of stating performance is of general interest as it furnishes a common basis of comparison for all heat engines.

4. The **Rankine cycle efficiency ratio,** which is the ratio of the thermal efficiency of the actual turbine to that of a theoretical turbine operating on the Rankine cycle.

This ratio is of particular interest to the designer, for it indicates to what extent he has succeeded in reducing the thermal losses.

Performance figures are most easily expressed on the basis of electrical output, but, if the generator efficiency is known, they may readily be calculated to the brake- or shaft-horsepower basis. Strictly speaking, the Rankine cycle ratio should always be calculated on the basis of the output of the turbine only, but it is often impracticable to determine the generator losses and the ratio must then be calculated on the basis of generator output.

In order to compute these performance results for a turbogenerator, only four measurements are required, *viz.*, steam quantity, inlet-steam conditions, exhaust pressure, and electrical output. The inlet-steam conditions of pressure, temperature, and quality are easily determined, but the difficulties encountered in accurately determining the other quantities are sufficient to make the test interesting. Each of these measurements is discussed in Sec. IV, page 411.

As already suggested, acceptance tests on the main turbogenerator units in a commercial plant are of considerable importance both to the seller and to the purchaser of the units, especially if a bonus and penalty clause is included in the contract. In order that the results of the tests should be fully acceptable to both parties, it is necessary that the details of procedure and calculation be fully agreed upon before the tests are started. Among the points that require settlement, the following are important:

1. The particular instruments to be used for each observation and how these instruments are to be calibrated.
2. Specific details regarding the methods to be used in determining the total steam, the load, the condenser leakage, and, if operating condensing, the absolute exhaust pressure or vacuum.
3. Methods to be used for determining corrections for variations in initial steam pressure, superheat or quality, exhaust pressure, steam extraction, and other operating conditions.
4. Conditions of inaccuracy or inconsistency of results which are to be recognized as sufficient to justify the rejection of a test run.

II. PREPARATIONS

1. Instruments and Apparatus.—The apparatus to be used will depend upon the conditions of the test. Throttle, inlet, and stage pressures above atmosphere will be measured with calibrated indicating Bourdon gages, throttle and extraction temperatures with calibrated indicating mercury thermometers, steam quality with a throttling calorimeter, speed with a high-speed counter or a calibrated tachometer, exhaust pressure with a mercury-column manometer, and atmospheric pressure with a mercury barometer. Barometers and mercury-column readings should be corrected for temperature (58.4°F standard), and for differences in elevation (see Chap. II, page 19). Recording gages and thermometers are usually not sufficiently accurate for test work. The steam consumption should be determined by measuring the weight or the volume of the condensate. The electrical meters for load measurement should be calibrated, and, if watt-hour meters are used, the revolutions of the disk as well as the readings of the integrating dials should be observed. For a bleeder turbine, the extracted steam should be condensed and weighed.

2. Special Preparations.—The following special preparations may be necessary, particularly when testing a large machine:

a. If the machine is to operate under a commercial load, special provisions must be made before the test so that the machine may be given a "steady" load.

b. The boiler operators must be notified when the test is to be run and requested to hold boiler pressures and loads as steady as possible.

c. A special inspection should be made of all by-pass lines, tie lines, drains, etc., through which leakage might occur that would affect the results of the test.

d. Provisions should be made for determining and holding constant the water level in the condenser hot well.

e. Condenser leakage tests should be provided for.

f. Provisions should be made to take into account the quantity of condensate, if any, used for sealing turbine glands, pump glands, atmospheric relief valve, etc.

g. Steam escaping from the steam seals or labyrinth packing should be measured, and, if end-tightened labyrinth packing is used, the axial position of the spindle should be observed.

h. For a bleeder turbine, provision should be made so that the vents from the extraction heaters do not blow an excessive amount of steam.

III. INSTRUCTIONS

1. Determinations.—Since turbine guarantees are usually made at one-half, three-quarters, and full rated load, and sometimes at one and one-quarter load (25 per cent overload), it is desirable that tests be made at these points. For turbines having a secondary or overload valve (operated either by the governor or by hand), it is important that a test be made at the maximum load that the machine will carry when the overload valve is closed.

Although it is possible, by plotting a Willans line, to closely approximate the performance curve of a turbine from two or three runs, the test curve can hardly be considered accurate unless four or more determinations have been plotted. In addition to the loads above mentioned, a light-load or no-load run should be made. (See instructor regarding loads to be used in the present test.)

It is always desirable to make a preliminary run and to calculate the results of this run before starting the formal test. The methods and conditions for the preliminary run should correspond with those that are to be used in the formal test. The A.S.M.E. Test Code[1] gives the following instructions for checking the constancy of operating conditions:

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[1] In addition to the A.S.M.E. Code, there is an international standard Code on steam-turbine testing, known as the "Rules for Acceptance Tests" of the International Electrotechnical Commission.

During the period preliminary to a test run, readings shall begin after the load and other operating conditions have been held practically constant for about 15 min. Steam rate shall be computed for successive periods of from 3 to 15 min (such periods need not be of equal duration), and the formal run shall be deemed to have started with the first two such successive periods for which the uncorrected steam rate checks within 3 per cent. This checking of successive periods should be continued throughout the run, as an indication of the constancy of operating conditions.

During all runs, special attention must be paid to maintaining those conditions of throttle pressure and temperature, exhaust pressure and extraction, for which the turbine was designed or which are specified in the purchase contract.

The test Code specifies that each constant-load run be at least 1 hr long when condensate is weighed or measured (or at least 10 hr long when boiler feed is weighed or measured).

Special Test for Stage Efficiencies.—If high initial superheat, or reheat between stages, is available, it is possible to determine the heat content of the steam at the various stages by simple pressure and temperature measurements. Make such a test if possible, and plot the expansion line on the Mollier diagram. Plot also the straight-line expansion curve as determined from the power and steam-consumption data, and discuss the comparison of engine efficiencies (efficiency ratios) thus obtained.

2. Results and Calculations.—A suggested tabulation of data and results is given herewith as Form 80. Most of the items in this tabulation have already been explained, and certain others are discussed under Sec. IV, page 411. Items 20, 22, and 23 are defined on pages 400 to 402. Plot curves as directed on page 410.

The "corrected" steam consumption called for in item 11 forms the basis for the calculation of all the final performance figures. It is obtained by correcting item 10 for variations in throttle pressure and temperature, exhaust pressure, and extraction. "Correction curves" for making these corrections are usually furnished by the turbine manufacturer, and in acceptance tests they should be agreed to by both parties before the test is run. Sometimes auxiliary tests are run to determine the corrections. If correction curves are not available, the corrections may be approximated as follows:

Corrections for Variations in Throttle Pressure.—The variation of steam rate with pressure depends on the initial pressure, the back pressure, and the turbine load. The magnitude of the correction cannot therefore be specified in simple terms, except to say that it ranges from ½ to 3 per cent for a variation of 10 lb in initial steam pressure. A fair approximation can be made by obtaining the difference in adiabatic heat drop from initial pressure to back pressure, for a constant initial quality or degree of superheat. If for instance the total heat drop is 300 Btu and the difference in heat drop for a 10-lb difference in initial pressure is 3 Btu, the pressure

Form 80.—Results of Turbogenerator Test

Test of.....................Turbogenerator located at..................
Type of turbine.............Number of stages.....Type of governor.....
Type of generator...............Rated volts......Amps......Kw......
Conducted by..............Barometer...................................

	Run number	1	2	3	4
1	Duration of run, min				
2	Throttle pressure, gage				
3	Throttle temperature or quality				
4	Inlet pressure, gage				
5	Vacuum at exhaust, in. Hg				
6	Exhaust pressure, in. Hg abs				
7	Exhaust steam temperature, deg. fahr				
8	Total condensate, lb				
9	Total dry steam, lb				
10	Dry steam per hr., lb				
11	Dry steam per hr., corrected				
12	Revolutions per min				
13	Amperes				
14	Volts				
15	Kilowatts				
16	Efficiency of generator, per cent				
17	B. hp				
18	Dry steam per kw-hr., lb				
19	Dry steam per b. hp.-hr., lb				
20	Thermal efficiency, per cent				
21	B.t.u. consumed per kw-hr				
22	Rankine-cycle efficiency, per cent				
23	Rankine-cycle efficiency ratio, per cent				

Date................ Observers:.

correction would be approximately 1 per cent. For light loads, the actual correction would be somewhat less than this.

Corrections for Variations in Superheat and Quality.—The effect of variations of superheat on the steam rate of turbines is about 1 per cent for every 10° of superheat, owing largely to the reduced fluid friction with superheated steam. For temperatures near saturation (say within 50°) the superheat correction is somewhat larger, and for high superheats the correction is somewhat smaller than above stated. In testing turbines that normally operate at moderate or high degrees of superheat, the variation of steam rate is usually very close to 0.8 per cent for each 10 deg of superheat, it being slightly larger for Curtis and slightly smaller for Parsons machines.

The advantages of superheated steam for use in a turbine are so great that few machines are operated on saturated steam. When it is necessary to test a machine using wet steam, the steam consumption is usually figured on the dry-steam basis, and no further correction for moisture is attempted.

Corrections for Variations in Exhaust Pressure.—One of the important advantages of the steam turbine over the steam engine is that it can be used to expand the steam to very low absolute pressures. An inspection of the heat-entropy diagram shows that in the high-vacuum range, a pressure change of 1 in. of mercury represents a heat drop of perhaps 10 per cent of the total heat drop from inlet to exhaust. It would therefore be expected that a small change in back pressure would have a marked effect on the steam rate. On account of the magnitude and the uncertainty of steam-rate corrections for variation in vacuum, tests should not ordinarily be made at exhaust pressures differing greatly from the normal or design standard. If the variation is only a few tenths of an inch of mercury, the correction may be approximated from the following:

TABLE XXXI.—VACUUM CORRECTIONS FOR STEAM TURBINES

Around 29 in. the steam rate changes about 0.8 per cent for 0.1 in. change in exhaust pressure

Around 28 in. the steam rate changes about 0.7 per cent for 0.1 in. change in exhaust pressure

Around 27 in. the steam rate changes about 0.6 per cent for 0.1 in. change in exhaust pressure

Around 26 in. the steam rate changes about 0.5 per cent for 0.1 in. change in exhaust pressure

Interpretation of Results.—The results of a steam-turbine economy test can best be interpreted by plotting a Willans line, *i.e.*, a curve of total steam per hour (item 11), against load (item 17 or item 15). Since a turbine is governed by throttling, the curve will consist of one or more straight lines. If the entire load range of the machine is cared for by one throttle valve admitting steam to the first stage, the Willans line will be straight. If heavy loads or overloads are cared for by valves admitting steam to larger nozzles, or to stages beyond the first, then the

Willans line will show a break (or increase of slope) at the load at which the additional valves were opened. Inconsistencies in the test results are much more easily found by plotting the Willans line than by plotting a steam-rate curve. In fact, the best way to find the steam-rate curve is to plot it from the completed Willans line, using 10 or more points, equally spaced on the line itself and to plot the test points (item 18 or 19) *after* the steam-rate curve has been drawn in. This method of plotting is specified in the A.S.M.E. Test Code.

Additional curves should be plotted, showing the inlet or steam-chest pressure, the thermal efficiency, and the Rankine cycle efficiency ratio, plotting against load as before. These curves may be plotted direct from the test data (items 4, 20, and 23).

TABLE XXXII.—ENGINE EFFICIENCIES FOR TYPICAL MACHINES
(Ratio of actual thermal efficiency to efficiency of an ideal Rankine cycle; based on steam entering the throttle valve and on power measured as per rating)

Type and service	Rated output	Engine efficiency	
		Rated load	Half load
Turbines:			
Central station turbo-alternators; condensing; non-extraction.	20,000 gen. kw.	75	69
	1,000 gen. kw.	70	62
Turbo-generators for isolated stations; condensing.	750 gen. kw.	67	58
	100 gen. kw.	48	40
Turbines for driving pumps, fans, small generators, etc.; non-condensing.	250 shaft hp.	50	42
	75 shaft hp.	42	35
	50 shaft hp.	39	33
	25 shaft hp.	35	29
Engines:			
High-grade engine: uniflow, poppet valve, or Corliss; condensing.	500 shaft hp.	70	70
	150 shaft hp.	65	65
Slide-valve engine; cut-off governed; condensing.	50 shaft hp.	45	40
Same; non-condensing.................	50 shaft hp.	60	55
Small, throttling-governed engine; non-condensing.	25 shaft hp.	45	38

The test results should be compared with the results of tests on similar units and these comparisons discussed in the conclusions. The only satisfactory basis for comparison is the Rankine cycle efficiency ratio, since this is the only performance unit that gives comparative values in spite of variations in throttle pressure and temperature, and in exhaust pressure. In Table XXXII are presented some typical values for the performance of steam engines and turbines.

IV. NOTES AND PRECAUTIONS

1. Measurement of Steam Quantity.—The steam consumption should preferably be determined by condensing the exhaust steam and measuring its weight or volume. If the machine and the boilers serving it can be isolated, it is permissible to measure the water fed to the boilers, taking due precautions against leakage between the boiler inlet and the turbine throttle. The test Code states that the latter method "should give results accurate to within plus or minus 3 per cent."

Sometimes it is possible to remove and calibrate the first-stage nozzle block, *i.e.*, to measure the flow through these nozzles for various values of the inlet pressure. During such a calibration, the steam should be condensed and weighed, using any suitable surface condenser so arranged that the back pressure on the nozzles does not exceed 58 per cent of the absolute initial pressure.

Although the measurement of turbine condensate or boiler feed by meter is not recognized by the test Code, this method must sometimes be resorted to. If either a venturi meter or a V-notch weir meter is used, and the meters are carefully calibrated, the accuracy of this method is fairly satisfactory.

Whatever method of measurement is used, precautions must be taken that the quantity of water measured actually corresponds to the quantity of steam passing the turbine throttle and that the steam entering the throttle is dry or superheated. The test Code states that, "the moisture content of wet steam has a disproportionately large effect on the turbine steam rate and every effort should be made to avoid testing any turbine with a wet-steam supply." Usually the greatest source of difficulty from leakage is due to leakage of condenser cooling water into the condensate stream. The importance of checking this leakage is indicated by the fact that the A.S.M.E. has issued a 17-page test Code on the subject (Instruments and Apparatus, Part 21, Chap. I).

The steam space of the condenser may be filled with water before a test and the tube sheets inspected for leakage. The test Code suggests that the purity of the condensate be checked every half hour during the test by electrical-conductivity measurement, or, when using salt water, by a silver nitrate titration. Other sources of leakage which should be considered are steam-sealed and water-sealed glands on the turbine, sealing water used for the hot-well pump and the atmospheric relief valve, steam-driven oil pump, and vents or other vacuum lines leading to or from the condenser shell. If the make-up is admitted at the condenser hot well, it should be cut off during the test. When a bleeder turbine is being tested, special provisions must be made for accurate measurement of the extracted steam.

2. Measurement of Exhaust Pressure (see page 420).

3. Measurement of Power Output.—In addition to the points mentioned under Preparations, page 405, it should be noted that any line losses between the generator and the switchboard are a part of the generator output. The power required for field excitation of an alternating-current generator may or may not be furnished by the turbine, but, if it is, it is usually considered as one of the generator losses. In any event, some agreement should be made before the test, as to whether or not it is to be measured and how it is to be charged. For methods of power measurement when the turbine does not drive an electric generator, see Chap. II, page 98.

4. Significance of Inlet and Stage Pressures.—If previous tests of the same machine are available, a comparison of the observed inlet and stage pressures with former values will serve as an index of the internal condition of the turbine. The entire turbine or any part of it may be considered as a large steam-orifice meter. For a given condition of the turbine, the inlet pressure or any stage pressure plotted against load will give a straight line. Comparing the lines so obtained on previous tests with those obtained from recent observations will serve to indicate any changes in the turbine resulting from such causes as serious erosion or loss of blading, clogging by foreign matter, or bending of the nozzle walls.

5. Exhaust-steam Temperature.—When a small turbine operates on highly superheated steam, the exhaust is likely to be superheated also, and, in testing such a machine, this superheat temperature should be carefully measured. In fact, it is well to observe exhaust-steam temperatures in any turbine test, though these readings should not be used as a basis for the determination of exhaust pressure.

6. Correction for Gland Sealing.—Compensation for steam rate for condensate used for sealing off glands should be made.

7. Steam-line Leakages.—All lines leading from and to condenser other than the air and condensate pump should be blanked off.

EXPERIMENT 81. Test of a Surface Condenser

I. PREFACE

A complete test of a condenser of any type involves the performance of auxiliary units for furnishing the cooling water, and for removing the air and the condensate, and also the performance of the main unit exhausting into the condenser. In the present experiment we shall limit our attention to surface condensers and shall consider the performance of the auxiliaries only so far as such performance directly affects the condenser performance.

Condenser tests are usually made with full load on the turbine or engine, since condenser performance always improves as the load on the condenser is decreased. Three variables remain to be considered as regards their effect on the absolute pressure at the inlet of the condenser, *viz.:* (1) the quantity of circulating water, (2) the temperature of the circulating water, and (3) the amount of air entering the condenser. Although the temperature of the circulating water varies with the season of the year, it is usually practically constant for the length of a test period. In the present experiment, therefore, two sets of runs are called for, one in which the air leakage is varied and one in which the quantity of cooling water is varied. It is not possible to hold all other conditions strictly constant, but the small variations encountered may to some extent be taken care of in the calculation of results.

II. PREPARATIONS

1. Apparatus and Setup.—Although there are a number of precautions (see page 419) to be observed in a condenser test, the readings to be taken are few, and no unusual apparatus is required. Thermometers should be placed to indicate the temperatures of the circulating water entering and leaving the condenser, the temperature of the steam entering and of the condensate leaving, and the temperature of the air discharged from the air pump. The ranges of the thermometers should be such that readings to 0.1°F may be estimated between the scale divisions, and all thermometers should be checked for accuracy before the test is started. Usually a check by comparing the instruments with each other at operating temperatures is sufficient. The vacuum at the condenser neck should be measured by an accurate 30-in. mercury-column manometer. Means should be provided for weighing the steam condensed, for weighing or metering the cooling water, and for metering the air discharged from the air pump.

2. Special Preparations.—Before running a test on a condenser, every effort should be made to eliminate leaks, (see Experiment 80, Notes and Precautions, page 411). When a condenser is in operation, the pressure in the steam space is of course very much lower than the pressure outside the shell or that within the tubes or in the water boxes. Leakage of either air or circulating water into the steam space is highly objectionable. To stop air leaks, all joints subjected to a vacuum on one side and atmospheric pressure on the other should be painted with heavy asphaltum paint while the condenser is in operation. The A.S.M.E. Code presents the following methods for detecting water leakage:

a. Electrolytic conductance, where the cooling water contains salt or, if fresh, contains more than 350 parts per million of dissolved solids.

b. Silver nitrate titration, where the cooling water is salty.

c. Direct weight, where the cooling water is fresh, or (by agreement) with salt water.

The following methods may also be used in checking leakage:

1. Drain the condensate from condenser and then fill the steam space with water. An inspection of all joints will reveal any major air leaks. Inspection of the tube sheets will reveal water leaks.
2. An approximate rate of water leakage into the condensing space may be ascertained by measuring the water collected at the condenser hot well, when the steam is cut off and the circulating pumps are in operation.
3. An indication of the amount of air leakage may be obtained by noting the rate of pressure increase in the condenser after suddenly cutting off both the steam supply and the air pump.
4. Large air leaks may be located with the aid of a candle flame.

In the case of large condensers a small amount of water leakage may persist, and this can be corrected for by measuring its rate as directed below and subtracting this amount from the weight of condensate.

For the air-leakage test prescribed below (Form 81*c*), some means must be provided for controlling the admission of air into the exhaust steam before it reaches the condenser inlet.

III. INSTRUCTIONS

1. Determinations. *a. Effect of Air Leakage.*—Starting with the maximum vacuum obtainable, increase the pressure for each succeeding run, in steps of about 1 in. of mercury until four runs have been made, and thereafter in steps of 2 in. of mercury until the air valve is open wide. The runs themselves need be only as long as is necessary to secure reasonably accurate readings of the quantities of condensate, air, and cooling water. The other readings are to be taken at 2-min intervals, and the individual runs are not to be started until three consecutive sets of temperature and pressure readings show a close check. In other words, ample time must be allowed for conditions to reach an equilibrium, after a change has been made in the air-valve setting.

b. Effect of Quantity of Circulating Water.—At least four runs should be made, beginning with the maximum amount of cooling water available and reducing the amount of water supplied for each subsequent run until the temperature rise of the cooling water has been increased to about three times its initial value. Ample time must be allowed for temperature equilibrium to be reached in each case, before the readings are started.

2. Results and Calculations.—It should always be kept in mind that the prime function of a condenser is to reduce the exhaust pressure (and temperature) of the steam engine or turbine exhausting into it. It is desirable that a condenser produce the required vacuum with the least possible amount of cooling water of a given temperature and at least expense for investment in tube surface and shell. Other desirable features are that the pressure drop through the condenser shall not be large and that the condensate coming from the hot well shall not be much below the saturation temperature corresponding to the pressure of the entering steam. The performance of the condenser in the above respects is indicated by the various items in the tabulations called for in this experiment (Forms 81*a*, *b*, and *c*).

Form 81*a*.—Surface-condenser Test—General Data

Condenser made by.. . . ... . .Serial No . ...
Condensing steam from .. . .
Shape and over-all dimensions of shell . .
Number of tubes . . .. Length of tubes . Gage of tubes ... .
Total active condensing surface, external area of tubes between sheets (only). ...
Material of tubes . .. .
Kind of packing .. ..
Date and method of last cleaning. . . . .
Cleanliness of tubes .. ..
Source of cooling water . .. .
Minimum temperature of cooling water in winter . . . .
Maximum temperature of cooling water in summer .. ..
Condenser guarantee: . ...
....
....

Date of test.............. Observers: .

Form 81*b*. Surface-condenser Test—Effect of Air Leakage

	Item	Units	Run No. 1	2	3	4
1	Duration of run	Min.				
2	Observed barometer	In. Hg				
3	Corrected barometer	In. Hg				
4	Observed vacuum at inlet	In. Hg				
5	Corrected vacuum at inlet	In. Hg				
6	Absolute pressure at inlet	In. Hg				
7	Observed steam temperature	Deg. fahr.				
8	Steam pressure corresponding to observed steam temperature	In. Hg				
9	Temperature cooling water at inlet	Deg. fahr.				
10	Temperature cooling water at outlet	Deg. fahr.				
11	Temperature condensate from hot well	Deg. fahr.				
12	Temperature rise of cooling water	Deg. fahr.				
13	Terminal difference	Deg. fahr.				
14	Hotwell temperature depression	Deg. fahr.				
15	Arithmetical mean temperature difference	Deg. fahr.				
16	Logarithmic mean temperature difference	Deg. fahr.				
17	Load on engine or turbine	Hp. (or kw.)				
18	Steam condensed per hr	Lb.				
19	Cooling water per hr	Lb.				
20	Heat transfer based on item 15 per sq. ft. cooling surface per hr. per deg.	B.t.u.				
21	Heat transfer based on item 16 per sq. ft. cooling surface per hr. per deg.	B.t.u.				
22	Air discharged by vacuum pump (at 60 deg. fahr. and 14.7 lb.)	C.f.m.				
23	Air in condenser by weight, measured	Per cent				
24	Air in condenser by weight, calculated	Per cent				

The **terminal difference** is the difference between the outlet temperature of the cooling water and the saturation temperature corresponding to the pressure of the entering steam. If a condenser is able to operate with a small terminal difference for a given vacuum, good design is indicated. A small terminal difference means that a lower pressure can be maintained for a given cooling-water (inlet) temperature, or that a smaller quantity of cooling water is necessary than when the terminal difference is large.

Form 81c.—Surface-condenser Test—Effect of Quantity of Circulating Water

Item number	Item	Unit	Run No			
			1	2	3	4
1	Duration of run	Min				
2	Corrected barometer	In. Hg				
3	Corrected vacuum at inlet	In. Hg				
4	Absolute pressure at inlet	In. Hg				
5	Temperature cooling water at inlet	°F				
6	Temperature cooling water at outlet	°F				
7	Temperature condensate from hot well	°F				
8	Temperature rise of cooling water	°F				
9	Terminal difference	°F				
10	Hot-well temperature depression	°F				
11	Arithmetical mean temperature difference	°F				
12	Logarithmic mean temperature difference	°F				
13	Load on engine or turbine	Hp (or kw)				
14	Steam condensed per hr	Lb				
15	Cooling water per hr	Lb				
16	Average water velocity through tubes	Ft per sec				
17	Heat transfer based on arithmetic mtd, per sq ft per hr per deg	Btu				
18	Heat transfer based on logarithmic mtd, per sq ft per hr per deg	Btu				
19	Air discharged by vacuum pump (at 60°F and 14.7 lb)	Cfm				
20	Air in condenser by wt, measured	Per cent				
21	Steam condensed per hr per sq ft of surface	Lb				
22	Circulating water per hr per sq ft of surface	Lb				
23	Circulating water per lb of steam condensed	Lb				

The hot-well temperature depression, undercooling, or "hot-well refrigeration," as it is sometimes called, is the difference between the temperature of the condensate leaving the hot well and the saturation temperature corresponding to the pressure of the entering steam. For a given steam pressure, it is desirable that this hot-well temperature depression be small, as a low condensate temperature calls for just so much more heat to be supplied by the feed-water heaters.

The transfer of heat from the exhaust steam to the cooling water in a condenser is dependent upon the temperature difference, or difference in "heat head," existing between the two fluids. Since this temperature difference varies in different parts of the condenser, it is necessary to arrive at some method of calculating its average value, if an over-all coefficient of heat transfer from the steam to the water is to be arrived at (see Chap. V, page 206). In condenser practice two methods of calculating the "mean temperature difference" are recognized. The **arithmetical mean** temperature difference is obtained by simply subtracting the average of the cooling-water inlet and outlet temperatures from the condenser-steam temperature as observed. The A.S.M.E. Code calls for the use of the **logarithmic mean** temperature difference, defined as follows:

$$\text{log mtd} = \frac{t_2 - t_1}{\log_e \dfrac{t_s - t_1}{t_s - t_2}}, \tag{78}$$

where t_s is the observed steam temperature, and t_1 and t_2 are respectively the inlet and the outlet temperatures of the circulating water (see derivation, page 209).

The percentage of air in the condenser by weight may be calculated by applying Dalton's law of partial pressures, and the calculated value serves to check the measurements (see items 23 and 24, Form 81*c*).

Let T = observed condenser-inlet temperature, degrees Fahrenheit absolute,

P_s = saturation pressure corresponding to observed condenser-inlet temperature (from steam tables), pounds per square foot,

P_c = observed condenser-inlet pressure, pounds per square foot,

P_a = partial pressure of the air in the condenser, pounds per square foot,

V_s = specific volume of dry saturated steam at the pressure P_s

= volume of 1 lb of steam in the steam-air mixture,

V_a = specific volume of dry air at pressure P_a
= volume of 1 lb of air in the steam-air mixture,
Q = volume of the condenser-steam space, cubic feet.

By Dalton's law

$$P_a = P_c - P_s.$$

From the perfect-gas law,

$$P_a V_a = RT,$$

$$V_a = \frac{RT}{P_c - P_s}.$$

Since density is the reciprocal of specific volume, the total weight of the steam-air mixture contained in the space Q may be expressed as

$$W = \frac{Q}{V_s} + \frac{Q}{V_a}. \tag{79}$$

The calculated percentage of air by weight in the steam-air mixture is then (see item 24, Form 81*b*):

$$A = \frac{Q/V_a}{\dfrac{Q}{V_s} + \dfrac{Q}{V_a}} \times 100 = \frac{V_s}{\dfrac{RT}{P_c - P_s} + V_s} \times 100. \tag{80}$$

This calculated percentage of air by weight will differ somewhat from that obtained from the measurements of air and condensate, as the water entrained in the exhaust steam is neglected in the above method of calculation.

IV. NOTES AND PRECAUTIONS

1. Temperature Measurements.—Some difficulty may be experienced in obtaining the true temperature of the exhaust steam entering the condenser. In the case of large units, in particular, the readings of thermometers placed in various parts of the stream may not agree. Care should be taken that thermometer bulbs are located as far as possible from the metal parts of the condenser, and special extended stem thermometers should be used, if necessary. It is well to use two or more thermometers, and the readings should be averaged unless specific errors in the readings can be located. In a large circulating-water discharge pipe there may be some tendency toward stratification in the water stream. The A.S.M.E. Test Code specifies that "not less than three thermometers uniformly spaced" shall be used to obtain the average cooling-water outlet temperature. This precaution, however, is hardly necessary in a small installation.

2. Inlet Pressure.—Absolute pressures in the range of 0.5 to 1.5 in. of mercury are difficult to measure with accuracy. In fact, it is easier to measure much lower pressures, because for such work special instruments (such as the McLeod gage) are available. With the ordinary barometer and the mercury-column manometer an error of 0.02 in. in the reading of each instrument is not uncommon, and this may result in an error of 4 per cent in the absolute pressure when the pressure is around 1 in. abs. Errors of this magnitude may be expected, even when great care is taken. It is true that an "absolute-pressure gage" may be constructed, which combines the functions of the barometer and the manometer. This instrument (which is nothing more than a short barometer tube partly filled with mercury and formed into the shape of a U) finds favor with some engineers. But it is subject to large errors if not properly made and properly handled, so that the test Code and also the instrument makers do not recommend its use in power-plant tests. In order that the ordinary manometer and barometer readings may be as accurate as possible, the instruments used should be readable with accuracy to at least 0.01 in. with a vernier slide and some arrangement to avoid parallax. The zero mercury levels of both instruments should be checked both before and *after* the test readings. If there are indications of impurities in the manometer tube, it should be thoroughly cleaned and refilled with fresh or filtered mercury. The barometer should always be checked against a U. S. Weather Bureau instrument. The method of connecting manometers to the condenser is important. The test Code specifies that each manometer shall be mounted within 6 ft of the pressure hole in the casing and that it must be as nearly as possible free from vibration. The Code requires, further, that the pressure hole shall be normal to the inner wall, that it shall not be smaller than ½ in. standard iron-pipe size, and that there shall be approximately one such hole and gage for each 16 sq ft of inlet-conduit area but in no ordinary case less than 2 or more than 8. The connecting piping should slope toward the condenser, if possible; it should be free from water pockets, and a water trap with a drain should be provided near the manometer.

3. Water-pressure Measurements.—Differential mercury manometers shall be used to measure the friction or drop in pressure between inlet and outlet water boxes. A discrepancy of more than 1 in. of mercury between readings shall not be allowed.

4. Measurement of Air Leakage.—With a rotary or a jet type of air pump, the air leakage is usually best measured by means of a series of orifices or nozzles. The air pump may discharge into an equalizing chamber or drum and some arrangement be made whereby nozzles of various sizes can be placed in the discharge from this drum. A small water manometer may be used to read the back pressure in the drum. This pressure should be kept within the range of about 1 to 6 in. of water. The equations and coefficients of discharge given in Chap. VII, page 263, are sufficiently accurate, and a special calibration of the device is unnecessary. When a reciprocating vacuum pump is used, a gasometer is the best measuring device, though orifices may be used if large receivers are inserted to damp out the pulsations (see page 301). Care should be taken that the operating conditions of the air pump (jet pressures or speed) do not change while a run is in progress.

5. General.—The temperature of the circulating water leaving the condenser is normally 10 to 20°F lower than the temperature corresponding to the steam pressure in the condenser.

The weight of circulating water per pound of steam condensed is 25 to 50 lb with low vacuum and 60 to 120 lb for vacuums over 27 in. based on a 30-in. barometer.

Every effort should be made to conduct test run in accordance with specified conditions in order to reduce to the minimum the necessity for correction factors that may be of questionable accuracy.

CHAPTER X

REFRIGERATION AND COOLING

PREFACE

A refrigeration machine is a **heat pump** for the purpose of absorbing heat at a low temperature level and discharging it at a high temperature level. The object is usually to cool materials to a low temperature. A few installations are being made for all-year operation, using the **reversed refrigeration cycle** for winter heating, as well as the **direct refrigeration** cycle for summer cooling.

An entirely different method, by which moderate reductions in temperature are obtained, is that of **spray cooling.** By spraying water into dry air, both the water and the air tend to assume the wet-bulb temperature, according to the principle of *adiabatic saturation* (see Experiment 55, page 247). The use of this process is of course limited to applications in which cooling to the wet-bulb temperature is sufficient for the purpose in hand.

The capacity of a refrigerating system is expressed either in Btu per hour or in "tons." A **standard ton of refrigeration** is equivalent to 12,000 Btu per hr (or 200 Btu per min) of refrigerating effect. In 24 hr this amounts to 288,000 Btu, which is equivalent to the latent heat of fusion of 2,000 lb of water ice.

Compression refrigeration machines are the most common type. In these machines a *vapor refrigerant* is used, which absorbs heat by boiling in an **evaporator** at a low saturation temperature and discharges the heat in a **condenser** at a high saturation temperature. A **compressor** is used to change the saturation pressure (and temperature) from the low level to the high level. The condenser and the piping between the compressor and the **expansion valve** are known as the *high side.* The evaporator and the piping between the expansion valve and the compressor are known as the *low side* (see Fig. 189). Condensed tables of the properties of the two most common refrigerants, ammonia (NH_3) and freon (CCl_2F_2), are presented in Table XXXIII.

Brine and other antifreeze solutions are used for transferring the refrigerating effect from the evaporator to storage rooms or other points of use. Table XXXIV gives the properties of the four most common antifreeze solutions.

Table XXXIII.—Properties of Refrigerants

Saturation temp, °F	Ammonia				Dichlorodifluoromethane (Freon 12)			
	Pressure, abs	Sp vol vapor	Enthalpy above −40°F		Pressure, abs	Sp vol vapor	Enthalpy above −40°F	
			Liquid	Vapor			Liquid	Vapor
0	30.42	9.116	42.9	611.8	23.87	1.637	8.25	78.21
5	34.27	8.150	48.3	612.1	26.51	1.485	9.32	78.79
10	38.51	7.304	53.8	614.9	29.35	1.351	10.39	79.36
15	43.14	6.562	59.2	616.3	32.44	1.230	11.48	79.96
20	48.21	5.910	64.7	617.8	35.75	1.121	12.55	80.49
25	53.73	5.334	70.2	619.1	39.32	1.020	13.66	81.06
30	59.74	4.825	75.7	620.5	43.16	0.936	14.76	81.61
35	66.26	4.373	81.2	621.7	47.28	0.863	15.88	82.16
40	73.32	3.971	86.8	623.0	51.68	0.792	17.00	82.71
45	80.96	3.614	92.3	624.1	56.38	0.730	18.12	83.25
50	89.19	3.294	97.9	625.2	61.39	0.673	19.27	83.78
55	98.06	3.008	103.5	626.3	66.73	0.620	20.40	84.30
60	107.6	2.751	109.2	627.3	72.41	0.575	21.57	84.82
65	117.8	2.520	114.8	628.2	78.48	0.532	22.72	85.32
70	128.1	2.312	120.5	629.1	84.82	0.493	23.90	85.82
75	140.5	2.125	126.2	629.9	91.10	0.457	25.08	86.32
80	153.0	1.955	132.0	630.7	98.76	0.425	26.28	86.80
85	166.4	1.801	137.8	631.4	106.3	0.395	27.48	87.28
90	180.6	1.661	143.5	632.0	114.3	0.368	28.70	87.74
95	195.8	1.534	149.4	632.6	122.7	0.343	29.93	88.19
100	211.9	1.419	155.2	633.0	131.6	0.319	31.16	88.62

For volume of superheated vapor, use Charles' law.
For enthalpy of superheated vapor use specific heats as follows:
Ammonia: C_p varies from 0.55 at 0°F to 0.70 at 100°F.
Freon: C_p varies from 0.14 at 0°F to 0.18 at 100°F.

EXPERIMENT 82. Test of Ammonia-Compression Refrigerating Plant

I. PREFACE

The Test Code for Refrigerating Systems, published jointly by the A.S.R.E. and the A.S.M.E. specifies that the *standard rating of a refrigerating machine* using liquefiable vapor is the number of standard commercial tons of refrigeration it performs with a compressor-inlet pressure that corresponds to a saturation temperature of 5°F, and with a compressor-discharge pressure that corresponds to a saturation temperature of 86°F. These pressures are to be measured within 10 ft of the compressor.

TABLE XXXIV.—PROPERTIES OF ANTIFREEZE SOLUTIONS
Freezing point and percentage of antifreeze by weight

Freezing temp, °F	Calcium chloride brine		Sodium chloride brine		Alcohol and water		Ethylene glycol and water	
	% by weight	Specific gravity, 60°F	% by weight	Specific gravity, 60°F	% by weight	Specific gravity, 60°F	% by weight	Specific gravity, 60°F
25	7	1.06	6	1.04	9	0.982	15	1.015
20	11	1.10	9	1.07	15	0.970	17	1.017
15	14	1.12	13	1.10	19	0 962	21	1.021
10	16	1.14	16	1.13	22	0.955	25	1.025
5	18	1.16	19	1.15	26	0.948	29	1.029
0	20	1.18	21	1.16	29	0.942	33	1.033
−5	22	1.20			32	0.936	36	1.036
−10	23.5	1.21			35	0.930	39	1.039
−15	24.5	1.23			38	0.924	42	1.042
−20	25.5	1.24			40	0.919	45	1.045
−25	26.5	1.25			43	0.914	47	1.047
−30	27.5	1.26			45	0.910	49	1.049

For complete tables see A.S.R.E. "Refrigerating Data Book."
Specific heats vary as follows:
Calcium chloride: 0.85 with 10 per cent salt to 0.68 with 25 per cent salt.
Sodium chloride: 0.88 with 10 per cent salt to 0.81 with 20 per cent salt.
Alcohol: 0.99 with 15 per cent alcohol to 0.80 with 40 per cent alcohol.

The two most important results of a test of a refrigerating system are capacity and economy. Refrigerating capacity or output is expressed as "standard commercial tons of refrigeration produced." The economy is stated as "horsepower of compressor per standard commercial ton of refrigeration." This is based either on the indicated horsepower of the compressor or on the brake horsepower required to drive it. Usually both values are determined. The economy is sometimes stated in other terms also, depending largely on the type of driving unit.

Other important test results are the volumetric efficiency, the mechanical efficiency, and the compression efficiency of the compressor. These are defined the same as for an air compressor (see page 339). The heat transfer in the condenser and in the evaporating coils, and the general heat balance are also calculated in a complete test. The "coefficient of performance" is the ratio of refrigerating effect to compressor horsepower (both expressed in Btu per hour). It measures useful effect per unit of power cost and is therefore favored by many engineers, but it is not recognized by the Code.

II. PREPARATIONS

Instruments and Setup.—Suitable pressure gages and thermometers should be obtained and placed as shown in Fig. 188.

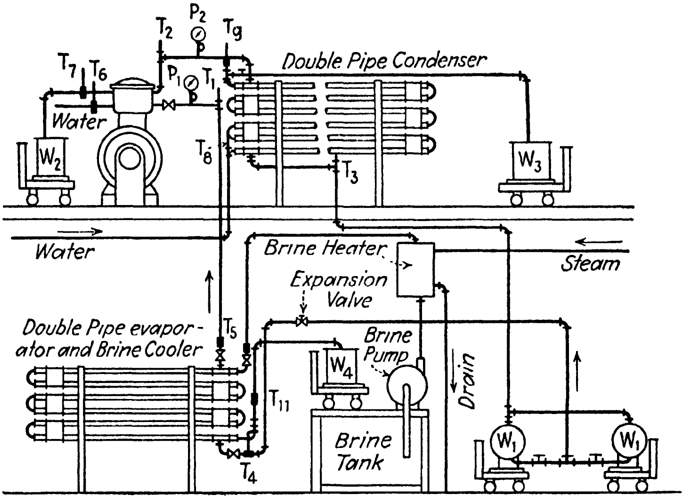

FIG. 188.—Apparatus for test of refrigerating plant.

Other equipment includes scales and weights, ammonia indicators and cards, counter, brine hydrometer, data boards and carbon sheets.

Although the setup of Fig. 188 is typical, several variations are possible. Instead of weighing the ammonia, it may be measured by volume, using two vertical tanks of small diameter and operating them alternately in the same manner as when weighing. Shell-and-tube evaporators are common, in place of the double-pipe unit. A closed brine system with electric heating is an improvement over the open system with steam heat, shown in Fig. 188.

If the driving motor or prime mover is to be tested, the instructor will furnish additional instructions.

III. INSTRUCTIONS

1. Determinations.—The entire test is to be run at constant conditions and the conditions specified in the test code should be duplicated as closely as possible (see paragraph 2, below). The duration of the test should preferably be at least 12 hr. The test code gives the following formula for the minimum duration of a test:

$$tw_1 = 3w_2, \tag{81}$$

where t = length of test, hours.
w_1 = rate of circulating refrigerant, pounds per hour.
w_2 = total weight of refrigerant contained in system.

2. Starting and Stopping.—Before starting the test, the plant should be operated a sufficient time so that the temperature and pressure conditions have become constant.

The saturation temperatures given in the Code (see "Preface," above) correspond to compressor-inlet and discharge pressures of 34.3 and 169.2 lb per sq in. abs, respectively. The Code further specifies that "there shall be 9 deg fahr subcooling of the liquid entering the expansion valve and 9 deg fahr superheating of the vapor entering the compressor." These conditions should be established as soon as possible after the system has been put into operation. They are regulated as follows: Discharge pressure is reduced by opening the condenser water valve; suction or low-side pressure is reduced by closing the expansion valve; evaporator-outlet temperature is reduced by closing brine valve or reducing the heater temperature; compressor-discharge temperature is reduced by opening compressor jacket-water valve. The liquid-ammonia receivers and weighing tanks will each be equipped with gage glasses, and the level in each should be the same at the end as at the beginning of the test. These levels should be recorded when the test is started.

Precaution.—Ammonia gage glasses are dangerous; do not touch them.

3. Data and Results.—Forms 82*a*, 82*b*, and 82*c* are suggested for the tabulation of data and results. The symbols used on Form 82*b* are the same as those used on the sketch. In calculating the results called for on Form 82*c*, the following suggestions will be helpful:

Item 1.—The available cooling effect is calculated from the weight of refrigerant circulated and the heat contents of the refrigerant entering and leaving the evaporator.

Items 2 *and* 3.—Item 2 is calculated from item 1, and item 3 is a similar quantity calculated from the weight, temperatures, and specific heat of the brine.

Item 5.—The gross horsepower is the indicated horsepower of the power cylinders in the case of a steam or internal-combustion-engine-driven compressor or the brake horsepower delivered to the compressor shaft in the case of a motor, belt, or gear-driven compressor.

Form 82*a*.—Test of Ammonia-compression Refrigerating System

Sheet 1—Description and dimensions

Plant location..
Type of service..
Compressor built by..
Type and rating of driving unit..
Capacity in tons per 24 hr......Method of volume control.............
Type of valves......Rated speed......Rated discharge pressure........
Cylinder diameter........Stroke........Piston-rod diameter...........
Rated displacement......Clearance in per cent of displacement.........
Diameter of intake pipe..........Diameter of discharge pipe...........
Type of condenser..
Area of condenser surface on refrigerant side..........................
Type of evaporator..
Area of evaporator surface on refrigerant side..........................
Composition of brine..
Average specific gravity of brine......Average specific heat of brine......
Type and rating of brine pumps..
Source of cooling water..
Type and rating of water pumps..

Date of test.......... Observers:...................................
....
....

Items 8 *and* 9.—The theoretical horsepower for adiabatic compression is obtained from the ammonia tables and charts. The measured quantity of ammonia is assumed to be compressed from the pressure and temperature observed at the compressor inlet, along a constant entropy adiabatic to the observed discharge pressure.

Items 10, 11, *and* 12.—Same as for an air compressor (see Experiment 70, page 339).

Items 13 *and* 17 are calculated from the weights, temperatures, and specific heats of the condensing water and the brine, respectively.

Items 14 *and* 18 are similar quantities, calculated from the weights and heat contents of the refrigerant.

Items 15, 16, 19, *and* 20.—Same as for a surface condenser (see Experiment 81, page 418).

Item 21.—Coefficient of performance is available cooling effect (item 1) divided by the heat equivalent of the indicated horsepower.

The heat balance serves as a general check on the experimental results by offering a comparison of the total heat absorbed with

Form 82*b*.—Test of Ammonia-compression Refrigerating System
Sheet 2—Test Readings

Item number	Item	Symbol	Observer number	Units	Average readings
1	Duration of test..................	...		Hr.	
	Average pressures:				
2	Compressor intake, gage........	P_1	1	Lb. per sq. in.	
3	Compressor intake, absolute....	....		Lb. per sq. in.	
4	Compressor discharge, gage.. ...	P_2	1	Lb. per sq. in.	
5	Compressor discharge, absolute..	....	..	Lb. per sq. in.	
6	Ammonia in condenser, gage ..	P_3	3	Lb. per sq. in.	
7	Ammonia in evaporator, gage ...	P_4	6	Lb. per sq. in.	
8	Brine-pump discharge............	P_5	6	Lb. per sq. in.	
	Average temperatures:				
9	Ammonia vapor, compressor intake ..	T_1	1	Deg. fahr.	
10	Ammonia vapor, compressor discharge	T_2	1	Deg. fahr.	
11	Ammonia liquid, condenser outlet. .	T_3	3	Deg. fahr.	
12	Ammonia mixture, evaporator inlet ..	T_4	6	Deg. fahr.	
13	Ammonia vapor, evaporator outlet...	T_5	6	Deg. fahr.	
14	Compressor jacket-water inlet......	T_6	1	Deg. fahr.	
15	Compressor jacket-water outlet	T_7	1	Deg. fahr.	
16	Condenser cooling-water inlet.	T_8	3	Deg. fahr.	
17	Condenser cooling-water outlet......	T_9	3	Deg. fahr.	
18	Brine at evaporator inlet...........	T_{10}	6	Deg. fahr.	
19	Brine at evaporator outlet..........	T_{11}	6	Deg. fahr.	
	Unit quantities:				
20	Liquid ammonia to evaporator	W_1	..	Lb. per hr.	
21	Compressor jacket water............	W_2	..	Lb. per hr.	
22	Condenser cooling water............	W_3	..	Lb. per hr.	
23	Brine circulated................	W_4	..	Lb. per hr.	
24	Compressor speed.............	....	.	R.p.m.	

Form 82c.—Test of Ammonia-compression Refrigerating System
Sheet 3—Calculated results

		Units	Results
	Refrigerating output or capacity:		
1	Available cooling effect	B.t.u. per hr.	
2	Refrigeration produced	Tons	
3	Refrigeration produced (brine method)	Tons	
	Power:		
4	Total i. hp. of compressor	Hp.	
5	Gross or b. hp. to drive compressor	Hp.	
	Economy results:		
6	I.hp. per ton of refrigeration	Hp.	
7	Gross or b. hp. per ton of refrigeration	Hp.	
	Efficiency results:		
8	Total theoretical hp. for adiabatic compression	Hp.	
9	Theoretical hp. per ton, adiabatic compression	Hp.	
10	Compression efficiency	Per cent	
11	Volumetric efficiency	Per cent	
12	Mechanical efficiency of compression	Per cent	
	Condenser performance:		
13	Heat removed by condensing water	B.t.u. per hr.	
14	Heat removed as calculated from refrigerant condensed	B.t.u. per hr.	
15	Logarithmic mean temperature difference	Deg. fahr.	
16	Coefficient of heat transfer based on logarithmic mean temperature difference	B.t.u.	
	Evaporator performance:		
17	Heat removed from brine	B.t.u. per hr.	
18	Heat removed as calculated from refrigerant evaporated	B.t.u. per hr.	
19	Logarithmic mean temperature difference	Deg. fahr.	
20	Coefficient of heat transfer based on logarithmic mean temperature difference	B.t.u.	
21	Coefficient of performance		

the total heat rejected. A tabular heat balance should therefore be given in the report. The items are as follows:

Heat balance:

1. Heat added in evaporator.
2. Heat added in piping.
3. Heat added as work (indicated horsepower).
4. Total heat added.
5. Heat rejected in condenser.
6. Heat rejected to compressor jacket water.
7. Heat lost from pipes and receivers.
8. Total heat rejected.

The report should also include an ammonia temperature-entropy diagram of the entire cycle, plotted to scale.

IV. NOTES AND PRECAUTIONS

Special all-steel indicators and steel thermometer wells must be used for ammonia, since ammonia attacks brass. If it is necessary to use a brass indicator for a few readings, it must be washed immediately with a weak acid solution such as vinegar.

The specific heats and densities of calcium and sodium chloride brines vary with the salt content. Values are given in Table XXXIV.

The lubricating oil used in the compressor cylinders must be of the special grade usually known as "ice-machine oil" (see Table L, page 490).

The test code sanctions the use of certain types of meters for measuring the refrigerant and states that calibrated volumetric (vertical) receivers may also be used. For determining the quantities of water and brine, the venturi or the orifice may be used if the precautions for installation and use given in Chap. VII, page 263, are followed.

It is well to protect observers by placing a strip of shatterproof glass in front of each ammonia gage glass. Scales for measuring height of liquid should not touch the gage glasses. Liquid heights may be marked by attaching spring paper clips (clamps) or clothespins to the scales and moving these to a new setting for each reading.

For low-clearance, power-driven compressors, such as are usually installed in engineering laboratories, all three of the test efficiencies will probably be near 80 per cent (items 10, 11, 12, Form 82*c*), and the brake horsepower per ton of cooling effect will be about 1.5 (item 7) (see Table XXVIII and Fig. 190 for additional performance values). These values apply to the operating conditions specified in the code. Heat-transfer coefficients vary greatly with conditions, but average values are given in Table XVIII, page 20[illegible]

EXPERIMENT 83. Test of a Small Condensing Unit

I. PREFACE

One of the outstanding contributions of the mechanical engineer in recent years has been the quantity production of small refrigerating machines. Not only for domestic refrigeration, but also for commercial establishments, the small **condensing unit,** such as that shown in Fig. 189, has become an indispensable part of modern living. Powered by electric motors, in sizes from ¼ to 10 hp (or more), these self-contained units operate under automatic control to furnish continuous cooling for everything from frozen fish and ice cream to drinking water coolers and air conditioning for living rooms.

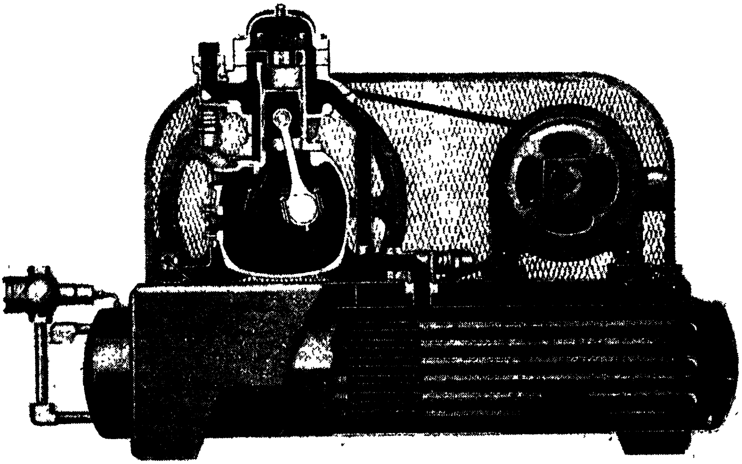

Fig. 189.—Sectional view of twin-cylinder condensing unit with fin-tube condenser inside storage reservoir. (*General Electric Company.*)

The present experiment is a test of a condensing unit, operating in connection with one or more evaporators or direct expansion coils. As far as convenient, the test methods will be those prescribed by the "Standard Method of Rating and Testing Mechanical Condensing Units," which is the standard Code, sponsored by the A.S.R.E. This Code specifies that the evaporator shall be arranged as a calorimeter, but for the purposes of a demonstration test, the use of direct expansion coils delivering refrigeration to either water or air is quite satisfactory. A secondary or confirming test is to be run simultaneously with the calorimeter test, according to the Code. In the confirming test the heat quantities are obtained either by measuring the quantity of refrigerant or by measuring the quantity and the temperatures of the condenser cooling water.

The **test results** to be obtained are: (1) Power input in horsepower or watts. (2) Capacity rating in tons. (3) The "performance factor" expressed in tons per kilowatt. (4) Cooling water consumption in gallons per hour. There are four standard groups of operating conditions, depending on the service for which the unit is intended. In all cases the condenser is to operate with an entering vapor temperature of 65°F and an entering temperature of cooling water of 75°F. The four standard evaporator (saturation) temperatures are: -10°F, $+5$°F, $+20$°F, and $+40$°F. For the lowest evaporator temperature, the cooling water to the condenser should leave at 85°, and for the highest evaporator temperature the cooling water should leave at 95°. In the other cases the cooling water leaves at 90°F. An ambient temperature of 90° is specified for all tests.

II. INSTRUCTIONS

1. Preparations.—The instruments and equipment required are essentially the same as shown in Fig. 188 for the test of an ammonia plant, but the methods will be modified on account of the compactness of the equipment. Some of the temperatures may conveniently be taken with thermocouples, using small wire soldered to the tubing or fittings, and covered with insulation. In order that the evaporator may function as a calorimeter, all heat loss to the air must be avoided by thoroughly insulating the cold surfaces.

2. Determinations.—Tests should preferably be made at one or more of the standard operating conditions specified in the Code. The test is essentially a long-time run at constant conditions, but some time will be required to attain the temperatures prescribed. To facilitate accurate temperature adjustments, all readings should be taken every 5 min throughout the adjustment period as well as during the test.

3. Results and Report.—In addition to the four test results already mentioned, the report should include a summary of all text data and a brief description of the unit, with name-plate ratings, compressor size and displacement, and operating speed. The Code requires elaborate calibrations of the evaporator calorimeter to correct for heat-transfer losses. If it has been impossible to use a calorimeter or to cover all cold surfaces with adequate insulation, corrections should be applied by calculating the heat absorption from the surroundings. (Measure surface areas and temperatures and use Table XVII, page 204.) At the higher evaporator temperatures, the performance factor should be slightly above 1 ton per kilowatt, for units of 3 hp or larger. The approximate effect of evaporator temperature and size of unit on the performance may be noted from the curves of Fig. 190.

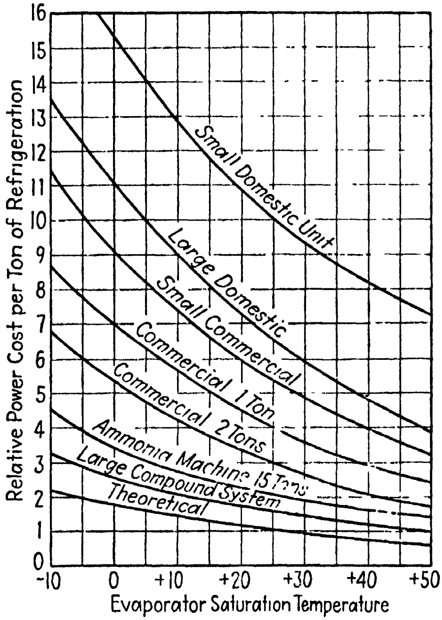

FIG. 190.—Relative power requirements of refrigerating machines. (*Refrigerating Data Book.*)

CHAPTER XI

INTERNAL-COMBUSTION ENGINES

I. INSTRUCTION

The internal-combustion engine is made in such a wide variety of sizes and types that a general study of its performance characteristics leads into several specialized fields of engineering. If we list a number of the major applications or uses of the internal-combustion engine, as for instance the airplane, the motor ship, the automobile, the farm tractor, and the diesel stationary plant, it becomes at once apparent that each field of application has several important engineering problems of its own.

The aircraft engine must perform under wide variations of atmospheric pressure and temperature, and it must, though of very light construction and small size, be capable of operating for long periods at high speeds and heavy loads, with the utmost reliability. The automobile motor, on the other hand, operates most of the time at a small fraction of its rated power and at various speeds. Rapid acceleration and high maximum speed are considered by the average motorist of more importance than economy. Engines for buses, trucks, and tractors usually operate at moderate speeds, but these services demand very rugged, long-lived motors and in some instances these motors must use kerosene fuel. In the field of stationary engines, the requirements vary from the intermittent demand for 1 to 10 hp in the case of a single-cylinder farm engine or contractor's engine, to central-station service demanding an output of 500 to 5,000 hp and more per unit. The fuels used vary from motor gasoline to heavy fuel oil or light crude.

II. GENERAL INSTRUCTIONS

In testing an internal-combustion engine, the objects of the test will of course vary somewhat with the type of service for which the engine is to be used, but in most cases the two major results are thermal efficiency and mechanical efficiency. Sometimes these results are expressed in other terms, such as pounds of fuel per horsepower-hour, Btu per horsepower-hour, friction horsepower, etc., but in any case the essential measurements are the following:

1. Heating value of the fuel.
2. Rate of fuel consumption.
3. Engine speed.
4. Engine torque (or power output).
5. Mechanical friction (or difference between indicated horsepower and brake horsepower).

The most common test consists of several runs at the same speed but with different loads, runs being repeated at other speeds when a variable-speed engine is under test.

Although efficiency and economy are the most common results desired, the primary object of the test may be something quite different, such as a comparative study of different fuels, of air-fuel ratios, or of the effect of some design characteristic of the engine.

Each of the essential measurements listed above involves certain peculiarities not encountered in the testing of other kinds of machines, and these measurements will therefore be discussed briefly.

1. Heating Value of the Fuel.—The A.S.M.E. Test Code for Internal-combustion Engines makes the following recommendations regarding the determination of the heating value of the fuel (see also page 359):

> The "higher heating value" is to be used in all cases. For gaseous fuels, a standard form of gas calorimeter shall be used, and the same operated as one of the instruments of the test. For liquid fuels, the heating value shall be determined in a standard bomb calorimeter, by a recognized physical or chemical laboratory. Calculation of the heating value of a gaseous fuel may be resorted to if a previous agreement has been made by the parties to the test.

The heating values of gasoline and kerosene may be estimated by the Sherman and Kropff formula, if such a procedure is made the subject of previous agreement. The Sherman and Kropff formulas are as follows:

For gasoline: $H = 18{,}320 + 40\ (\text{deg A.P.I.} - 10)$. (82)

For kerosene: $H = 18{,}440 + 40\ (\text{deg A.P.I.} - 10)$. (83)

For distillate fuel oils: $H = 18{,}650 + 40\ (\text{deg A.P.I.} - 10)$. (84)

(See Experiment 36 for conversions.)
Calorific and other fuel tests are covered by the A.S.M.E. Test Code for Liquid Fuels.

2. Rate of Fuel Consumption.—Measurements of gaseous fuel "must be made with meters or by methods yielding the greatest possible accuracy consistent with availability and cost in the large sizes."[1] Positive-displacement meters are preferred, though rate-of-flow meters may be resorted to in the large sizes, if proper precautions are taken (see Chap. VII, page 263). Proportional meters and anemometer-type meters should not be used. Both the A.S.M.E. and the S.A.E. codes specify that liquid-fuel measurements are to be made by weight. The fuel tank is placed on sensitive platform scales and the runs are started and stopped with the falling of the scale beam, a stop watch being used for timing. The accuracy of this determination may be increased and the necessary length of run reduced by substituting a balance (such as a trip scale or a torsion balance) for the platform scales. A mercoid switch attached to the balance provides a simple means of using the weight measurement to actuate relays for speed and time determination. Volume measurements are not recognized by the codes, with the exception of those made by using a quantity of fuel from a tank and then restoring, with a weighed quantity of fuel, the level as previously determined by a hook gage or on a gage glass. Volume measurements by various methods that involve the principle of the burette, the graduate, or the glass pipette or volume bulb are quite generally used in several industrial and educational laboratories, however, and are favorably regarded by them. Flowmeters are also used for adjustment purposes and instantaneous reading where extreme accuracy is not required.

3. Engine Speed.—The choice of speed-measuring instruments will depend upon the engine speed to be measured and upon the character of the test. The A.S.M.E. Code recognizes the use of either continuous stroke counters, revolution counters with friction drive, or tachometers (see Chap. II, page 61). It specifies that continuous stroke counters shall be used only for slow-running engines, where a constant speed of considerable duration is maintained. It specifies further that, if tachometers of the centrifugal, liquid-pumping, or magnetic types are used, they must be calibrated, and also that they shall not be connected to the engine shaft by a belt or similar drive. The S.A.E. Code does not sanction the use of a tachometer in the testing of an automotive engine, except as an approximate check or as an indicator of variations in speed during the run. This Code states that "speed should invariably be taken from positively driven continuous counters which engage at the beginning of the run and disengage at the end." This specification evidently refers to a counter with an electromagnetic clutch. By using such a counter having the electric circuit completed by a mercoid switch on the fuel scales, very accurate results may be obtained. The same mercoid switch may be connected to a relay tripping a stop watch or to close the circuit on a synchronous electric timer.

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[1] A.S.M.E. Code.

4. Engine Torque or Power Output.—Any type of absorption dynamometer or brake may be used (see Chap. II, page 98), but for high-speed engines the test codes specify the electric-cradle type. The water-cooled magnetic or eddy-current brake is a very compact unit for high capacities, but of course it cannot be used as a motor. For airplane engines in particular it may be desirable to supplement the electric dynamometer with a water brake or a Prony brake. Special care must be taken in **balancing the dynamometer** and in obtaining the dead weight on the scales, if any. The S.A.E. Code specifies that the dynamometer shall be balanced by operating it as a motor, without load, first in one direction and then in the other, and finally checked by hanging a check weight from the dynamometer arm. Calibrated platform or beam scales, and not spring balances, are to be used for weighing the test loads, but spring balances may be inserted between the dynamometer arm and the beam scales, and they are useful when adjusting the load and for the purpose of absorbing shocks and vibrations. When the engine under test is direct connected to an electric generator, it is usually necessary to make a test of the unit as a whole. If it is not practicable to make an electrical test or analysis for the calibration of the generator, the tests are reported on the basis of the electric horsepower or kilowatt load (see Experiment 80, page 403).

5. Mechanical Friction.—There are essentially three ways of measuring the "friction horsepower" of an internal-combustion engine: (1) by difference between the brake horsepower and the indicated horsepower; (2) by calculating it from the no-load or "friction card" taken with the indicator; (3) by measuring the power required to turn the engine over with the fuel supply and ignition cut off. It is difficult to obtain consistent results by any of these methods, and several precautions are necessary.

The common engine indicator, when used on the internal-combustion engine, has very decided limitations, and as stated by the A.S.M.E. Code "in no case can an indicator be regarded as a precision instrument on an internal-combustion engine to the same degree as on a reciprocating steam engine." The following limitations are specified by this Code:

a. For engine speeds above 400 rpm, the indicator shall not be used for determining the cylinder mean effective pressure.

b. For engine speeds above 400 rpm, optical indicators may be used on single-cylinder engines for indicating valve adjustments or spark timing.

c. The indicator shall not be used for determining cylinder mean effective pressure on hit-and-miss governed engines.

d. At speeds below 400 rpm, the indicator must be rejected as a means for determining cylinder mean effective pressure, if the cards for a given constant load do not check within plus or minus 2 per cent. The test for constancy is made by taking at least 10 diagrams on one card and repeating this test at various odd intervals. The largest area must not differ from the smallest by more than 4 per cent of their arithmetical mean.

e. When the cylinder pressures are very high and the total card area is small, the results are likely to be in error. The rejection of such cards is left to the judgment of the test engineer.

f. A separate indicator shall be used on each end of each cylinder. (It is important that the indicator connections be as short as possible.)

Several "high-speed indicators" have recently been developed, and one of these of the mechanical type is described in Experiment 11, page 77. Piezoelectric indicators utilizing the oscillograph are widely used for research work, and are available commercially.[1]

The determination of friction horsepower by the "friction card" is seldom resorted to and is not recognized by either of the test codes.

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[1] For a comprehensive discussion, see K. J. DeJuhasz, "The Engine Indicator, Its Design and Theory, *Instruments*, June 1932, to July, 1933.

For automotive engines the friction horsepower is determined by the third method mentioned above, *i.e.*, measuring the power required to drive the engine with the fuel supply and the ignition cut off. The S.A.E. Code specifies that the friction test be made immediately after the brake-horsepower test, before the engine has cooled and with the throttle in the same position as for the corresponding brake-horsepower test. Compression relief cocks remain closed, and all accessories, such as generator and pumps, should be in operation. The friction horsepower, as determined by tests at a number of speeds, may be plotted against speed and intermediate values read from the curves, see Fig. 191.

A **general precaution** which must be observed in all internal-combustion-engine tests is that time must be allowed for a "steady state" to be reached after making a load or speed change, before the test readings are started. Jacket-water and lubricating-oil temperatures must be allowed to become constant. The A.S.M.E. Code specifies that in no case shall test observations for the first run of a test be started less than 1 hr after the operating conditions have been imposed, and that a similar equalizing period of at least 10 min shall be allowed before each subsequent run.

EXPERIMENT 84. Preliminary Engine Studies

I. PREFACE

Time is usually not available for obtaining the general data on an engine during the same period when a test is run. It is important to collect the general information required by the test codes, and if this information is available prior to the tests, it is easier to interpret the engine performance. Hence this experiment covers a general survey of laboratory equipment and the assembling of preliminary data on the engines to be tested later (in Experiments 87, 88, and 93,) or on similar engines assigned by the instructor.

II. INSTRUCTIONS

1. Procedure.—Make a list of the engines set up in the laboratory, giving name, type, rating, speed range, and possible uses. Make a separate list of special equipment and instruments available for the testing of internal-combustion engines.

Form 84*a*.—Automotive Engine—General Data

Engine name and model..
Manufactured by..................Type of service..................
General type..
Number of cylinders.....Bore.................Stroke.................
Total piston displacement.......Compression (clearance) volume........
Compression (volume) ratio........Compression pressure...............
Cooling system...
Fan diameter.....Number of blades....Ratio fan to engine speed.......
Weight of engine.................lb., including.........................
Carburetor, name and model................Nominal size.............
Method of heating carburetor or charge ....
Ignition, name and type.................Firing order..................
Spark-plug location........................Gap..........................
Control of spark advance...
Lubrication system, type...
Accessories attached during test......................................
Fuel, kind.............Baumé gravity.........Heating value..........
Lubricating oil, make and grade.......................................
Oil viscosity, Saybolt universal: At 100 deg. fahr.....At 210 deg. fahr....
Dynamometer, make and type...................Arm.............ft.
Room temperature.......Barometer...........Humidity..............

Date of test.......... Observers:..... ....
....
....

Form 84*b*.—Diesel Engine—General Data

Engine name and model..... .. . .
Manufactured by........ ....Type of service............ ....
General type...................... ... .Speed, rpm.......... . .
Number of cylinders............Bore..................Stroke ....
Total piston displacement.............. ......Compression volume.............
Compression (volume) ratio....................Compression pressure.............
Single or en bloc cylinders.............. .....Liners................ .. .. .
Cooling system............... ....
Piston type..................Length.................Rings............ ..
Crankshaft bearings: No.......................Size........ . . ...
Flywheel OD.........................Weight of engine................ .....
Type fuel injection............................Pressure.....................
Name and model of fuel pump........ ....
Injection valve location.......... ....
Injection timing, fixed or variable........... ....
Air injection pump...........................Pressure...... ....
Scavenging air pump.........................Pressure...... ....
Lubrication system.................................... ....
Method of starting........... ....
Accessories...
Fuel used.............................Oil used...........................
Dynamometer or load...

Obtain, by observation, the data required by Form 84*a*, covering one automotive engine. Obtain similar data for Form 84*b*, covering one diesel engine.

For the automotive engine, make an actual measurement of clearance if possible, by removing the cylinder head and filling the combustion chamber recess with oil from a graduate (with crank on dead center).

Measure the linear opening of each valve on one cylinder of the automotive engine, at intervals of 15-deg crank angle, over the entire cycle. Locate dead centers by the angular method (Experiment 26). Determine breaker-point gap, spark-plug gaps, and spark timing.

2. Report.—Present the lists and forms in orderly tabular arrangement. Plot a curve of linear valve lift (on ordinates) against crank angle for both valves. Indicate dead centers and spark timing on the same diagram. Calculate the maximum compression pressure for both the automotive and diesel engines, using $PV^{1.3}$. Calculate theoretical Otto and diesel cycle efficiencies by Eq. (85), page 457, and Eq. (86), page 463.

EXPERIMENT 85. Automotive Fuels—Gasoline-distillation Test

I. PREFACE

Although gasoline is almost the universal fuel for automotive engines, there are several kinds and grades of gasoline. The two common tests for distinguishing the character of a given sample are the gravity or density test and the distillation test.

Gasoline is produced by three methods: by straight refining or distillation of crude petroleum, by "cracking" the heavier fractions that result from straight distillation, and by condensing from crude natural gas a product known as "casing-head gasoline." Commercial gasoline is usually a blend of two or more of these products. Additional constituents may be added to this blend for special purposes, the two most common of these being tetraethyl lead and benzol.

The gravity test (see page 155) gives a rough index of the nature of the blend and serves particularly to distinguish between the "aviation" and the "motor" grades. The distillation test is a reasonably accurate index of the performance of the gasoline in a motor, especially with regard to starting and "warming-up" characteristics. The distillation curve is closely related to the "volatility" curve, the latter being the measure of evaporation in air, as in a carburetor.

Other gasoline tests occasionally used are those for gum content, for sulphur, and for the vapor pressure.

II. INSTRUCTIONS

1. Procedure.—The standard 100-cu-cm Engler flask, condenser, shield, and electric heater are used. Fill the condenser bath with cracked ice, and add enough water to cover the condenser tube. Keep plenty of ice in the bath at all times. Swab out the condenser tube with a piece of soft cloth attached to a cord or wire, to remove any liquid remaining from the previous test. Measure out into a graduated cylinder 100 cu cm of the gasoline to be tested. Transfer it at once to the Engler flask, being careful not to let any of the liquid get into the side arm of the flask. Insert the thermometer into a cork, and fit this tightly into the flask so that the thermometer will be in the middle of the neck and so that the lower end of the capillary is on a level with the bottom of the vapor-outlet tube at its junction with the neck of the flask. Place the 6- by 6-in. asbestos board with the 1¼-in. circular opening over the electric heater, and set the charged flask over the opening. The side arm or vapor tube should be inserted into the condenser tube, making a tight connection by means of a cork. This arm should not extend into the condenser tube less than 1 in. or more than 2 in. Place the graduated cylinder that was used to measure the gasoline at the outlet of the condenser tube in such a position that the condenser tube will extend into the graduate at least 1 in., but not below the 100-cu-cm mark. Do not dry the inside of the cylinder. Cover the top of the graduate with a piece of blotting paper that has been cut so as to fit the condenser tube tightly.

Apply heat at a uniform rate, so regulated that the first drop of condensate falls from the condenser in not less than 5 or more than 10 min. When this first drop falls, record the thermometer reading as the **initial boiling point.** The graduated cylinder should then be moved so that the end of the condenser tube touches the side of the cylinder. Regulate the heat so that the distillation proceeds at a uniform rate of not less than 4 or more than 5 cu cm per min. Check this rate several times during the run. *Record the thermometer reading when the level of the distillate reaches each* 10-*cu-cm mark on the graduated cylinder.*

After 90 cu cm have distilled, the heat may be increased so as to vaporize the "heavy ends," which have relatively high boiling points. However, no further increase of heat shall be applied after this adjustment. The 4- to 5-cu-cm rate can rarely be maintained from the 90 per cent point to the end of the distillation, but under no conditions should the time be more than 5 min from the 90 per cent point to the end point. The heating shall be continued until the thermometer reading reaches a maximum and starts to fall consistently. The highest observed temperature shall be recorded as the **end point.** Usually this point will be reached after the bottom of the flask has become dry.

The total volume of the distillate collected in the receiving graduate shall be recorded as the **recovery.** The cooled residue shall be poured from the flask into a small cylinder graduated in 0.1 cu cm and the measured volume recorded as **residue.** The difference between 100 cu cm and the sum of the recovery and the residue shall be calculated and recorded as **distillation loss.**

Repeat the distillation, using a second portion from the same gasoline sample. If proper care has been taken, duplicate results should not differ from each other by more than 6°F. Duplicate temperature readings at the various percentage points should not differ by more than an amount equivalent to 2 cu cm of distillate at each point.

Make the gravity test on another portion from the same gasoline sample, and correct the result to 60°F (see Instructions, page 156).

2. Results.—Plot distillation temperature (on the ordinates) against percentage of recovery. Give table of results and compare with the specifications given on page 492.

EXPERIMENT 86. Automotive Engine—Tune-up and Adjustments

I. PREFACE

Much specialized equipment has been developed for the service-testing of automotive engines. But most of the important checking may be done with simple equipment, including a few electric meters and pressure gages, with suitable wiring and connectors.

Motor service-station tests may be divided into three classes, for checking respectively: (1) The general mechanical condition. (2) The electrical system and its parts. (3) The fuel system. Measurements of compression pressure and of the vacuum in the intake manifold give some idea of the condition of piston rings, cylinders, and valves. Electrical tests are concerned mainly with current and voltage readings on the low-tension side, and voltage and spark-gap tests on the high-tension side. Aside from a low battery or poor connections, ignition defects are most likely to be found in the spark plugs or the breaker points, although occasionally a poor condenser or a faulty coil will be located.

Performance of the fuel system is usually checked by means of the exhaust-gas analyzer, giving approximate readings of air-fuel ratio.

Typical values for test readings and adjustments are given in Form 86. These are not necessarily exact figures, since in most cases the performance will be acceptable over a range of values. When tests of each cylinder are made, as when measuring compression pressure or testing spark plugs, the uniformity of the readings is of more importance than their absolute value.

II. PREPARATIONS

1. Equipment.—The checking and "tuning up" of an automobile motor can be done with ordinary tools and instruments, but the special testers are of course more convenient.

2. Instructions.—The following list covers the simpler adjustments and the more important tests. If additional equipment is available, further tests will be outlined by the instructor.

A. Checking the general mechanical condition of engine:

1. Start engine and allow it to warm up, then connect vacuum gage to intake manifold. Note gage reading and fluctuation at idling speed. Also observe change in gage reading when throttle is suddenly closed (released) after high-speed operation.
2. Remove spark plugs and test compression in each cylinder with compression gage equipped with check valve. Recheck each cylinder after squirting in a little oil.

3. Clean and test spark plugs if equipment is available. Then adjust spark gaps of plugs and replace them.

B. Electrical checking—primary circuit:

1. Check the specific gravity of the battery solution.
2. Connect voltmeter across battery and check open-circuit voltage.
3. Connect voltmeter to generator lead and ground, and check generator voltage with engine running at about 30 mph. Check voltage when engine is turned by starter. Observe maximum generator charging rate on car ammeter.
4. Remove distributor cap and inspect all connections. Inspect breaker points and measure maximum breaker-point gap. Check condenser if equipment is available.
5. If equipment is available, take measurements of: (*a*) Primary voltage at coil. (*b*) Primary current at coil, engine running and not running.

C. Electrical checking—secondary circuit:

1. Make electrical tests according to equipment available for measuring: (*a*) High-tension voltage, at coil and at each spark plug, engine running 25 mph. (*b*) High-tension current, at coil and at each spark plug, engine running 25 mph.

D. Carburetor adjustment and gas analysis:

1. With vacuum gage connected to intake manifold, change idling adjustment on carburetor to obtain least fluctuation of gage.
2. Take readings with exhaust-gas analyzer, using sampling tube inserted in end of exhaust pipe. These readings should be taken at idling speed, medium speed, and during acceleration. Adjust carburetor as directed by instructor.

Use Form 86 for recording readings, and compare values with those given in table and with "Notes." Report should include a discussion of the results of each measurement and suggestions for improvement of the condition of engine and equipment.

III. NOTES AND PRECAUTIONS

1. If the gravity of all battery cells is at least 1.200 and the voltage at the generator is below 4.5 when starter is used, look for poor connections at battery terminals or defective cables.

2. If the voltage is below 7 with the engine running 30 mph, the generator or the cutout needs attention. If the voltage is above 9, look for a loose connection between generator and battery.

3. The current drawn by the starting motor should be between 100 and 250 amp.

4. Compression pressure should be nearly the same on all cylinders if valves and piston rings are in uniform condition. To distinguish between faulty ring action and faulty valve action squirt a little oil into the cylinder

Form 86.—Automotive Engine Adjustments

Car make and model...... .. ..Owner.................
Engine No................ .Mileage ... .Tested at....................

	Typical range of values	Observed
1. Intake-manifold vacuum, in. Hg		
a. Turned by starter......	6–8	
b. Engine idling	16–19	
c. Fluctuation when idling.	1–3	
2. Compression pressures, lb gage		
a. Dry.		
b. With oil		
3. Spark-plug gaps, in.	0.025–0.040	
4. Specific gravity of battery, min	1.200–1.250	
5. Primary voltage at battery:		
a. Engine not running	6	
b. Turned by starter	4.5–5.5	
6. Generator voltage, engine running at 30 mph. ...	7–8	
7. Generator charging rate, max		
8. Breaker-point gap, max, in..	0.020	
9. Primary voltage at coil		
10. Primary current to coil:		
a. Engine not running...	2–3	
b. Engine running..........		
11. High-tension voltage.....		
12. High-tension current.		
13. Air-fuel ratio		
a. Idling..............		
b. Accelerating......		
c. At 30 mph...........		
14. CO_2 in exhaust..................		

Date of test............... Observers:.

to help seal the rings. If there is no change in pressure the rings are sealing satisfactorily.

5. If all spark plugs do not test about the same after cleaning, check distributor cap and high-tension leads. If spark plugs are tested under pressure, defective plugs are more easily found.

6. Intake-manifold vacuum should be 6 to 8 in. of mercury when cranking with the starter, and 18 to 19 in. with the engine idling, with fairly steady readings in both cases. Low readings at steady idling speed may indicate retarded spark, and excessive fluctuations (more than 3 in.) are obtained with poor carburetor adjustment, a defective spark plug, or a leaky valve.

7. In the case of high-tension voltage and current readings *uniformity at all plugs* is of more importance than actual values or instrument readings. Primary current to coil should not be more than 3 amp at low speeds, and primary voltage should be from 4.5 at idling to 6 at medium speed.

8. Air-fuel ratio should be 11 or above, with not less than 9 per cent CO_2. If air-fuel ratio is 14 or above, the Orsat should be used for checking.

9. In making spark-gap settings or any other adjustments, follow the engine manufacturer's recommendations if they are available.

EXPERIMENT 87. Automotive Engine—S.A.E. Performance Test

I. PREFACE

Dynamometer or "block tests" of land-vehicle or aircraft engines are made for many purposes. In no other branch of the machinery industry is there to be found that unique combination of large-scale production and yearly models that is characteristic of automobile manufacture. Such a combination demands a most thorough experimental program, and hence we find that not only the engine manufacturer and the car builder but also the makers of carburetors, electrical equipment, and other accessories are engaged in a continuous program of engine testing. In many research problems concerned with engine fuels, lubri-

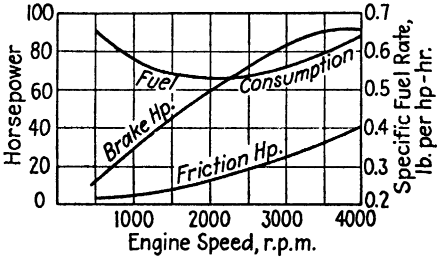

FIG. 191.—Power and economy curves for a typical automobile engine.

cation, cooling, and design, the dynamometer test furnishes the final verdict. Airplane engines are required to pass certain government standard dynamometer tests before being acceptable for use in a plane. One other type of test is sometimes made, *i.e.*, the "chassis dynamometer" test. This, however, is a test of a complete chassis assembly rather than one of the engine alone.

Since the test code of the Society of Automotive Engineers, which has already been quoted, is almost universally used for automotive-engine tests, the present test will be made to conform to its requirements as far as practicable. The curves of Fig. 191 show typical performance of an automobile engine of about 215 cu in. displacement.

II. PREPARATIONS

1. Apparatus.—The test code requires that an electric dynamometer be used. The dynamometer must be provided with calibrated platform or beam scales for load measurements and with a positively driven counter which may be engaged at the beginning of the run and disengaged at the end. The dynamometer must be checked for proper balance before the test. The fuel tank is supported on sensitive scales or balances, with a flexible rubber-hose connection or a siphon system to the carburetor. Mercury thermometers should be provided for measuring carburetor air-inlet temperature and inlet and outlet temperature of the cooling water. Other apparatus not called for by the test code but required to obtain data called for on Forms 87*a*, *b*, and *c* are a platform scales and tank for weighing the cooling water, a thermocouple pyrometer for obtaining the exhaust temperature, a Bourdon vacuum gage for obtaining the inlet-manifold pressure, a Bourdon pressure gage for measuring the oil pressure, a water (or mercury) manometer for measuring the exhaust back pressure, and a thermometer for obtaining the temperature of the lubricating oil.

2. Preliminaries.—Carburetor settings, breaker points, spark-plug gaps, etc., shall be adjusted before the test is started, and the control of the engine during the test shall be by spark and throttle only. Test runs should not be made until the engine has been thoroughly run in. As a test of this condition, no appreciable difference in friction horsepower should be shown before and after a 30-min run at the speed of maximum torque, with the throttle wide open. The fan, generator, and all other regular equipment shall be on the engine and operating during the test. Cooling-water outlet temperatures should be maintained at 175°F (±5 deg). An air draft should be provided (if practicable), which approximates that produced by the motion of the car on

Form 87*a*.—Test of Automotive Engine

(For Sheet 1 use Form 84*a*.)

Sheet 2—Brake horsepower and fuel consumption

Item number	Item	Units	Run No.			
			1	2	3	4
1	Duration of run..............	Min.				
2	Air inlet temperature.........	Deg. fahr.				
3	Cooling-water inlet temperature	Deg. fahr.				
4	Cooling-water outlet temperature.......................	Deg. fahr.				
5	Fuel temperature.............	Deg. fahr.				
6	Intake-manifold temperature..	Deg. fahr.				
7	Exhaust-gas temperature......	Deg. fahr.				
8	Lubricating-oil temperature....	Deg. fahr.				
9	Exhaust back pressure........	In. Hg				
10	Intake-manifold vacuum......	In. Hg.				
11	Lubricating-oil pressure.......	Lb. per sq. in.				
12	Weight on dynamometer scales	Lb.				
13	Torque......................	Lb-ft.				
14	Revolutions per minute.......					
15	Brake horsepower............	Hp.				
16	Brake m.e.p.................	Lb. per sq. in.				
17	Piston speed................	Ft. per min.				
18	Friction horsepower (from curve).....................	Hp.				
19	Indicated horsepower.........	Hp.				
20	Total fuel used..............	Lb.				
21	Fuel consumption per b. hp-hr.	Lb.				
22	Heat consumption per b. hp-hr.	B.t.u.				
23	Brake thermal efficiency.......	Per cent				
24	Heat lost to cooling water.....	Per cent				

the road at the speed corresponding to the engine speed. Before the first run is started, a warming-up period at the proposed test load, long enough for the establishment of a "steady state," must be allowed. The indications of this state are the constancy of cooling-water, oil, and exhaust temperatures. For further general instructions, see page 434.

Read the **operating precautions for electric dynamometers,** page 106, before starting the engine.

III. INSTRUCTIONS

1. Determinations.—Beginning with a run at wide-open throttle and maximum speed, additional runs should be made at various speeds down to the slowest steady operating speed of the engine, with the throttle wide open in each case. (Consult instructor regarding the maximum speeds to be used and the number of runs to be made between maximum and minimum speeds.) When this series of runs has been completed, a set of friction-horsepower runs should be made at the same speeds and with the same throttle setting. Before starting these runs the fuel supply is turned off and the carburetor run dry. The dynamometer is then immediately run as a motor to drive the engine, and the power measured.

Form 87*b*.—Test of Automotive Engine

Sheet 3—Friction-horsepower test

Item number	Item	Units	Run No.		
1	Duration of run..................	Min.			
2	Mean temperature of cooling water	Deg. fahr.			
3	Lubricating-oil temperature.......	Deg. fahr.			
4	Lubricating-oil pressure..........	Lb. per sq. in.			
5	Revolutions per minute..........				
6	Weight on dynamometer scales....	Lb.			
7	Friction horsepower.............	Hp.			

Another complete set of brake-horsepower and friction-horsepower runs is next made at throttle settings that allow the engine to develop, at each speed, three-fourths of the power that it developed with full-throttle opening at that same speed. Similar tests are made at one-half and one-quarter loads.

If time is not available for the complete test as outlined above, the three-quarter and one-quarter load tests may be omitted. The time required may be further abbreviated by assuming that the throttle setting at which the engine will develop half load at one speed will also be the half-load setting for all other speeds.

The duration of fuel-consumption tests shall not be less than 5 min and of friction-horsepower tests not less than 1 min.

Cold water must not be admitted to the engine jackets during a friction-horsepower run.

2. Calculations and Results.—Forms 87*a* and *b* are suggested for the presentation of data and results. Most of the items called for in these tabulations have already been discussed. The **brake mean effective pressure** (item 16, Form 87*a*) is calculated by using the familiar indicated-horsepower formula, horsepower = $PLAN/33{,}000$, substituting the brake horsepower as determined by test and solving for P. Account must be taken of the number of cylinders in the engine and the number of explosions per minute in each cylinder, *i.e.*, N is the number of power strokes per minute. The maximum brake mean effective pressure should be around 90.

The friction-horsepower values will be taken from the curves plotted as directed on page 439.

If the test results are to be compared with the results of tests made elsewhere, both sets of results should be **corrected to standard conditions** by making use of the relation which states that the power developed by an internal-combustion engine varies directly as the absolute barometric pressure and inversely as the square root of the absolute temperatures.

The S.A.E. Standard Curve Sheet provides that all test results are to be plotted against motor speed on the abscissa. The curves called for on this sheet are horsepower, torque, mechanical efficiency, fuel consumption, thermal efficiency, and brake mean effective pressure. These curves should be included in the report of the present test, a separate curve sheet being used for each throttle opening (full load, half load, etc.). Compare the test curves for full throttle with those given in Fig. 191.

EXPERIMENT 88. Stationary Engine—Economy Test

I. PREFACE

The stationary internal-combustion engine is usually a constant-speed machine operating under governor control, and the economy test is not much different from the similar test on a steam engine (see Experiment 79, page 398).

Although a multicylinder diesel engine is assumed in the discussion that follows, the general procedure will also apply to any stationary internal-combustion engine using gas, gasoline, or oil, and single cylinder or multicylinder.

II. PREPARATIONS

Apparatus.—The engine should be provided with a Prony brake or other suitable absorption dynamometer (see page 98) and also with indicator connections and an indicator reducing motion. The indicator should be of the special high-speed type, with small piston and light reciprocating parts.

Other apparatus required for the test is as follows: counter or calibrated tachometer, mercury thermometers for air-inlet temperature, for inlet and outlet temperature of the cooling water, and for temperature of the fuel; thermocouple pyrometer for exhaust-gas temperature; tanks and scales for weight of fuel and weight of cooling water; planimeters, and slide rules. Volumetric fuel and water measurements may be used if desired.

Before obtaining the apparatus and undertaking the test, the general instructions at the beginning of this chapter and the precautions (page 454) should be carefully noted.

III. INSTRUCTIONS

1. Determinations.—Four runs are to be made, at one-quarter, one-half, three-quarters, and full load, respectively. The A.S.M.E. Code provides that the duration of each run shall be at least great enough to ensure the accuracy of fuel measurement within 1 per cent. A ½-hr run with readings every 5 min will be sufficient in this case, provided care is taken that the six observations result in a set of uniform cards and that all conditions are held constant. If erratic readings appear, the observations should be continued until six consecutive readings

Form 88.—Test of Stationary Engine

Engine type and classification..
Engine made by..................Maker's rating.................
Number of cylinders....Cylinder diameter....Stroke...Rated speed....
Type of carburetor or fuel injection ....
Type of ignition ....
Baumé gravity of fuel used.........Higher heating value..............

Item number	Item	Units	Run No. 1	2	3	4
1	Duration of run....................	Min.				
2	Air-inlet temperature..............	Deg. fahr.				
3	Cooling-water inlet temperature......	Deg. fahr.				
4	Cooling-water outlet temperature.....	Deg. fahr.				
5	Fuel temperature..................	Deg. fahr.				
6	Exhaust-gas temperature............	Deg. fahr.				
7	Weight on brake scales, net..........	Lb.				
8	Torque..........................	Lb-ft.				
9	Revolutions per min..............					
10	Brake horsepower..................	Hp.				
11	Mean effective press. (from card).....	Lb./sq. in.				
12	Indicated horsepower..............	Hp.				
13	Friction horsepower...............	Hp.				
14	Horsepower by friction card.........	Hp.				
15	Mechanical efficiency..............	Per cent				
16	Total fuel used....................	Lb.				
17	Fuel consumption per b. hp-hr......	Lb.				
18	Brake thermal efficiency............	Per cent				
19	Indicated thermal efficiency.........	Per cent				
20	Heat lost to cooling water...........	Per cent				

Date................ Observers:

give a good check, and these six readings only will then constitute the run.

2. Results.—Form 88 indicates the results to be reported in this test. Curves should be plotted with brake-horsepower load on the abscissas, showing the variations of fuel consumption, mechanical efficiency, brake-thermal efficiency, and indicated-thermal efficiency. Comparisons should be made with Table XXXVI, items 1 and 2.

IV. SPECIAL INSTRUCTIONS AND PRECAUTIONS

1. Before starting the engine, see that the brake is anchored or held in such a way that it will not cause damage if the engine backfires.
2. Check the lubricating oil and the fuel supply.
3. Turn on lubricators and cooling water before (or *immediately* after) the engine has been started.
4. Keep the indicator stopcock closed except while taking a card. Leaving this cock even partly open for more than a very few explosions will result in inaccurate cards and damage to the instrument, owing to sticking of the indicator piston. The piston should be removed, wiped clean, and oiled a number of times during the test.
5. Take a "light-spring diagram" (with a 10-lb spring), to show the condition of the valve adjustments.
6. Be sure to allow sufficient time for the establishment of a "steady state" before beginning a set of readings (see page 439).
7. Indicator cards are not considered satisfactory if the extreme variation in diagram area is more than 4 per cent (see page 438).
8. Take a number of "friction cards" with the brake removed from the engine, and from these make a check determination of the friction horsepower.
9. Each student must mount one card from each run and also a friction card and a light-spring diagram in his report.

EXPERIMENT 89. Automotive Engine—Volumetric Efficiency

I. PREFACE

The operation of the vacuum windshield wiper and similar devices on an automobile indicates that the pressure in the intake manifold is well below atmospheric most of the time. Since this vacuum is produced by the downward-moving piston and there is some pressure loss through the intake valves, the cylinder pressure at the end of the intake stroke must be even less than that in the manifold. Thus the charge of air taken into the cylinder, being at a lower pressure and a higher temperature than that of the atmosphere, is much less dense. The ratio of the weight of air actually taken into the cylinders per minute to the weight of a similar volume of atmospheric air at the pressure and temperature existing near the carburetor inlet is called the "volumetric efficiency." This is not an "efficiency" in the usual engineering sense but rather a ratio of weights, densities, or volumes.

Although the volumetric efficiency has a minor effect on fuel economy, it affects greatly the total capacity or power developed and hence is watched most closely in the case of airplane engines, racing engines, or any engine in which the power-weight ratio is important.

The major factors that affect the full-throttle volumetric efficiency are the size and design of carburetors, intake manifolds and valves, the mixture temperature, and the engine speed. At partial throttle the volumetric efficiencies are of course still lower.

The most promising method for increasing volumetric efficiency is by supercharging.

II. INSTRUCTIONS

1. Preparations.—The significant results are obtained at high speed, and hence the engine should be prepared for a high-speed test, with a suitable means for measuring the intake air. An orifice or nozzle is the most convenient air meter, but the device must be carefully made and used, in accordance with the instructions given on pages 274 and 276, and special precautions are necessary to eliminate pulsations, as described on page 301. If a calibrated gasometer tank of sufficient size is available, this is of course to be preferred over the orifice method.

2. Determinations.—Make one run at full-open throttle and others at partial-throttle settings as designated by the instructor. Each run is to consist of five or more determinations at different speeds. Power and fuel should be measured as well as air flow.

3. Results.—Volumetric efficiency, brake horsepower, and fuel rate should be plotted against speed, using one curve sheet for each throttle setting. Volumetric efficiency should also be plotted against brake horsepower, one curve for each throttle setting.

III. NOTES

For tests dealing with volumetric efficiency, air-fuel ratio, etc., it is often desirable to simulate road-power conditions. For this purpose the power requirements given in Table XXXV, page 459, may be used.

EXPERIMENT 90. Automotive Engine—Air-fuel Ratio

I. PREFACE

The air-fuel ratio on an automotive engine is fixed by the carburetor design and the carburetor setting. The simple venturi-tube carburetor furnishes an increasingly richer mixture as the throttle is opened, and all automotive carburetors are provided with compensating devices in the form of auxiliary air valves, compensating jets, etc. Many present-day carburetors have only an idling adjustment, and the air-fuel ratio under load can be altered only by changing the fuel orifice or "jet."

Engine economy is increased as the ratio of air to fuel is increased, *i.e.*, as the mixture is made more "lean." High air-fuel ratios result in a lower maximum power and less rapid acceleration, as well as difficulty in starting. Since the latter qualities are readily detected, while economy is seldom accurately measured by the automobile driver, most carburetors are set for a richer mixture than the one for best economy. This is especially true in cold climates and in mountainous regions.

In this experiment, two methods of measuring the air-fuel ratio are described, but the same test may be run with either method separately.

II. INSTRUCTIONS

1. Preparations.—If possible the air-fuel ratio should be obtained both by computation from the exhaust-gas analysis as in Experiment 93, page 461, and by direct measurement of air and fuel (see Experiment 89).

2. Determinations.—With the throttle in a fixed position as designated by the instructor, make one set of runs with "normal" carburetor setting, one with a richer mixture, and one with a leaner mixture. Each set of runs is to include power, fuel, and air measurements at five or more different speeds.

3. Results.—Plot and discuss the results in the ways which you think best illustrate the variations of performance with air-fuel ratio. Compare exhaust-gas analyses with Table XLVIII, in Appendix.

EXPERIMENT 91. Automotive Engine—Compression Ratio

I. PREFACE

The compression ratio of the "average" American automobile engine has been increased more or less steadily ever since the first cars were produced. Gasoline has been improved and special fuels developed to allow higher compression pressures without "detonation" knock. Much research has been done in this field and in the study of the combustion process and of flame propagation, particularly in England and Germany.

The thermodynamic equation for the ideal Otto cycle states that the cycle efficiency is a function of compression ratio:

$$\text{Efficiency} = 1 - \left(\frac{V_2}{V_1}\right)^{k-1}. \qquad (85)$$

where V_2 = clearance volume.
V_1 = total cylinder volume.

The performance of an actual engine with different compression ratios may be demonstrated either by testing a conventional engine with two cylinder heads, or by using a special variable-compression or knock-testing engine. "High-compression" heads are obtainable for the popular automobile engines, and single-cylinder stationary engines are available with high- and low-clearance heads. The variable-compression engine is a special design on which the compression may be changed by shims under the head or by an adjustable barrel. The latter may be adjusted while the engine is in operation.

It may be desirable to change fuels as well as changing the engine-compression ratio.

II. INSTRUCTIONS

1. Determinations.—With the same engine, and with all settings of carburetor, throttle, and ignition the same in both cases, make two series of runs, one with a low compression ratio and the other with a high compression ratio. Take care that cooling conditions are maintained as nearly constant as possible throughout the tests. Obtain at least five runs in each series at various speeds within the range designated by the instructor.

Since the full compression pressure is available only at full-open throttle, the runs should preferably be made under these conditions.

If a special variable-compression or knock-testing engine is available, the standard procedure as recommended by the Cooperative Fuel Research Committee of the S.A.E. and the A.P.I. should be followed. This is a test of fuel properties rather than one of engine performance, and the fuel to be tested is compared with standard reference fuels consisting of iso-octane and normal heptane, mixed in various percentages. The percentage of octane in the mixture is called the "octane number." Secondary reference fuels which have been calibrated against the octane-heptane standards may be used for routine testing, since the standard fuels are very expensive.

The test consists of the comparison of the maximum engine knock produced by the gasoline under test with the knock produced by one higher and one lower reference fuel. Knock is measured by a "bouncing pin" or electric contact indicator, using either an electrolytic gas-evolution burette or an electrical resistance element and thermocouple. Ample time must be allowed after changing fuels for steady conditions to be established with the new fuel. Three alternate runs should be made on each fuel. Two fuels, five units apart, may be used and the rating of the test fuel determined by interpolation from knock meter or burette readings.

2. Report.—The nature of the results will depend on the equipment available. The report should be designed to illustrate best the effects of changing compression ratio and of changing fuels. Compare the compression ratios used with those that are standard for some of the well-known automotive engines.

EXPERIMENT 92. Automobile Economy Test

I. PREFACE

The results of automobile engine tests on the dynamometer block are not easy to interpret in terms of road performance because of the wide variety of engine operating conditions in ordinary driving. Actual road tests show the over-all economy under certain driving conditions, but full information on the car will call for a third series of tests, using a chassis dynamometer.

There are three general conditions of car operation under power: (1) Level-road conditions, (2) hill climbing, (3) acceleration. Table XXXV is a summary of the performance of an ordinary car on a level road, from 10 mph to maximum speed. These conditions of power and speed may be simulated in both a block test and a chassis dynamometer test. Thus the three types of tests may be correlated for level-road conditions. Similar correlations would be possible for hill climbing, though it is difficult to find uniform grades with different slopes, for road tests. Acceleration tests are the most difficult and can be duplicated in the laboratory only when the chassis dynamometer and the test block are equipped with heavy flywheel loads. The inertia of these loads must then be adjusted to simulate the inertia of the loaded car.

TABLE XXXV.—TYPICAL AUTOMOBILE PERFORMANCE*

For an "average" low-priced American car operating on a smooth concrete road at uniform speed, no stops

Car speed, mph	Power available		Power actually used			Economy	
	Max bhp	Net bhp (with fan and muffler)	Chassis friction, hp	Wind, resistance, hp	Total, hp	Lb fuel per net bhp-hr	Miles per gal
10	12	12	2	0	2	2.0	21
20	27	26	3	1	4	1.3	26
30	40	38	4	3	7	1.1	24
40	53	50	6	6	12	0 9	21
50	63	58	8	13	21	0.8	19
60	72	65	11	21	32	0.7	16
70	75	67	14	32	46	0.7	13
78	72	63	18	45	63	0.8	9

* Adapted from data by Zierer and Macauley, *J. S.A.E.*, January, 1939, p. 30.

NOTE: Ratio of engine speed to revolutions per minute of rear wheels in direct drive is usually about 4.0 to 1.

Even when road tests, chassis dynamometer tests, and engine block tests have all been made, for both level-road conditions, hill climbing, and acceleration, there are still many practical operating conditions that have not been covered. Extremes of winter and summer temperature, length of run, deceleration and traffic stops, and even the habits of the driver have their effect on the actual miles per gallon. All of these can be analyzed by the engineer, but the starting point in any given case is level-road operation.

II. INSTRUCTIONS

1. Preparations.—The same car should if possible be subjected to both a road test and a chassis dynamometer test. For most favorable results, the engine should be given the tune-up and adjustments described in Experiment 86, before the tests are started.

For mileage tests the burette type of fuel mileage tester, marked for accurate measurements of $\frac{1}{10}$ gal is most convenient. The commercial testers include a pump and a three-way valve, also rubber hose and fittings for connecting into the gasoline line on the suction side of the regular fuel pump.

Connect and mount the mileage tester for convenient operation by the observer in the front seat. Check all connections for leakage and fasten the rubber tubing at some distance from the exhaust pipe and manifold. Attach vacuum gage for intake-manifold pressure. Secure stop watch and data board and prepare data sheet.

2. Determinations. *Road Test.*—Select a level stretch of road about 3 miles long, and as free from traffic as possible. Make two tests at each speed, one in each direction, to cancel wind effects. Compute speed from readings of trip speedometer and stop watch. Plot miles per gallon and intake-manifold vacuum (on the ordinates), both against speed, and repeat any test points that do not lie on the smooth curves.

Chassis Dynamometer Test.—Make tests over the same speed range as used in the road tests. Adjust dynamometer load to produce the same manifold vacuum previously observed at the same speed. (If road tests on this car are not available, set dynamometer for total road power shown in Table XXXV.) Obtain exhaust-gas analyzer, insert sampling tube in exhaust line, and take readings during each test. Determine fuel mileage as in road test.

3. Report.—Performance curves should be presented and discussed in the report, with comparisons of results by road tests and chassis dynamometer. Discussion should include also a comparison with values given in Table XXXV, and with results of dynamometer tests on similar engines (see Fig. 191).

EXPERIMENT 93. Stationary Engine—Heat-balance Analysis

I. PREFACE

While the role of the diesel engine in the automotive field is not yet well established, it has certain advantages in the stationary field which cannot be questioned. For use in or near the oil-producing regions, for towns and small cities in sparsely populated districts, or for industrial plants and large buildings requiring power without a corresponding demand for heat, the use of the diesel engine is rapidly increasing.

The thermal efficiency of a diesel engine is not approached by steam plants of like capacity and is not equalled by even the best steam central stations. Since the diesel engine burns a much higher grade of fuel, however, the economic comparison may not be as favorable as the thermal.

In this experiment it is assumed that a thermal analysis is to be made of a stationary diesel engine, but a semidiesel or a gasoline or gas engine may be substituted if desired. The engine is to be operated at various loads under governor control. The most important results are the determination of air-fuel ratios, the heat-balance table, and the analysis of indicator diagrams (if indicator equipment is available). A typical diesel engine heat balance is given in Table XXXVI.

II. INSTRUCTIONS

1. Preparations.—From the list of heat-balance items given below, determine what readings and what instruments are necessary. Before starting the test, the relative performance of the various cylinders should be checked by comparing exhaust-gas

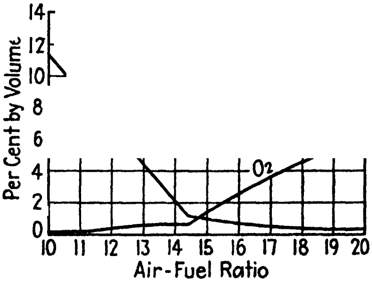

FIG. 192.—Orsat gas analyses for various air-fuel ratios, petroleum fuel.

temperatures at the individual cylinders. If large differences are found, consult the instructor regarding adjustments.

2. Determinations.—Make a complete test at each load, using preferably the full rated load, and one-fourth, one-half and three-fourths rated load. Tests must be long enough for the obtaining of several complete and consistent sets of Orsat analyses of the exhaust gases during each test, as well as accurate weights of fuel and cooling water. Check Orsat analyses by Fig. 192. Maintain the same cooling-water discharge temperature during all tests.

3. Results.—The items included in the engine heat balance are

a. Heat to useful work in brake horsepower.
b. Heat to work lost in engine friction.
c. Heat lost in the cooling water.
d. Sensible heat in the exhaust gases.
e. Hydrogen and moisture loss in the exhaust.
f. Losses due to incomplete combustion.
g. Radiation and unaccounted for.

TABLE XXXVI.—TYPICAL HEAT BALANCE FOR A SMALL DIESEL ENGINE

	Percentage of heat value of fuel			
Fraction of rated engine load	¼	½	¾	Full
1. Brake horsepower	24	29	30	30
2. Cooling water loss	34	31	29	28
3. Dry exhaust loss	21	21	23	24
4. Hydrogen-moisture loss	7	7	8	9
5. Radiation and unaccounted for	14	12	10	9

NOTE: Engine friction is dissipated as heat to cooling water, exhaust and radiation.

All values are given as percentage of the heat supplied in the fuel (higher heating value). Certain of these items may be combined if it is not practicable to determine them separately. The exhaust-gas analysis is made with the Orsat apparatus (see page 363 for instructions), and the calculations necessary for the determination of the various exhaust losses are practically the same as in the case of the boiler heat balance (see pages 387 to 390). When using either gasoline, kerosene, or light fuel oil, an ultimate analysis of 84 per cent carbon and 15 per cent hydrogen (balance neglected) may be assumed without introducing appreciable error into the heat-balance results. Heating values may be closely approximated from the formulas given on page 436.

Calculate the ideal engine efficiencies, and obtain the efficiency ratios, actual to ideal. The approximate cutoff ratio for each load on the actual diesel engine may be determined from the indicator cards, or, if cards are not available, consult the instructor for other suggestions. Calculate the air-fuel ratio by Eqs. (68) and (69), page 387, and check the results by Eq. (70) and by Table XLVIII in Appendix.

Plot curves of the heat-balance items against load (on the abscissa), using that scheme of plotting in which the curves are plotted one above another, and the distance between the final curve and the 100 per cent line represents the unaccounted-for losses. Plot also a curve of air-fuel ratio against load. Compare results with those given in Table XXXVI.

III. NOTES AND PRECAUTIONS

Allow ample time for obtaining steady conditions before starting each test and make observations frequently to check the constancy of conditions.

Fresh CO solution in the Orsat is important for exhaust-gas analyses.

Good performance for diesel engines lies in the range of 0.4 to 0.5 lb of fuel per bhp-hr at rated load, but the fuel rate of small engines is often between 0.5 and 0.7 lb per bhp-hr.

The methods of calculation already suggested make no allowance for the possibility of unburned hydrocarbons. This method is allowable with high air-fuel ratios, but, if the heat-balance analysis is made on a gas or gasoline engine, corrections for unburned hydrocarbons should be made.

The efficiency of the ideal diesel cycle may be calculated from the compression ratio r and the cut-off ratio s, as follows:

$$\text{Ideal efficiency} = 1 - \frac{1}{r^{k-1}}\left[\frac{s^k - 1}{k(s-1)}\right] \tag{86}$$

where k is the specific-heat ratio (1.4 for the "cold-air standard").

APPENDIX

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TABLE XXXVII.—CONVENTIONAL ABBREVIATIONS

Concept	Abbreviation	Concept	Abbreviation
Absolute	abs	Kilowatts	kw
Alternating current	a-c	Kilowatt-hours	kwhr
Amperes	amp	Linear	lin.
Area	spell out	Liters	spell out or l
Atmospheres	atm	Magnetomotive force	mmf
Boiler horsepower	boiler hp	Mean effective pressure	mep
Brake horsepower	bhp	Mercury	Hg
British thermal units	Btu	Meters	m
Calories	cal	Miles	spell out
Candlepower	cp	Miles per hour	mph
Centigrade	C	Milligrams	mg
Centimeters	cm	Millimeters	mm
Circular mils	cir mils	Millivolts	mv
Cubic	cu	Minutes	min
Cubic centimeters	cu cm	Number	No.
Cubic feet	cu ft	Ohms	spell out
Cubic feet per second	cfs	Ounces	oz
Degrees	deg	Page	p.
Degrees Fahrenheit	°F	Per centum	per cent or %
Diameter	diam	Pounds	lb
Direct current	d-c	Pounds per square inch	lb per sq in.
Electric horsepower	elec hp	Power factor	spell out or pf
Electromotive force	emf	Revolutions per minute	rpm
Fahrenheit	F	Revolutions per second	rps
Feet per second	fps	Saturated	sat
Feet	ft	Seconds	sec
Foot-pounds	ft-lb	Specific gravity	sp gr
Gallons	gal	Square	sq
Gallons per minute	gpm	Tons	spell out
Grams	g	Volume	Vol
Horsepower	hp	Watt-hours	whr
Hours	hr	Weight	wt
Inches	in.	Yards	yd
Indicated horsepower	ihp		
Indicated horsepower-hour	ihp-hr		
Kilograms	kg		
Kilogram-calories	kg-cal		
Kilometers	km		

See "Abbreviations for Scientific and Engineering Terms," American Standards Association.

For other forms and usages of technical English see "Style Manual for Engineering Authors and Editors," published by the A.S.M.E.

TABLE XXXVIII.—CIRCUMFERENCES AND AREAS OF CIRCLES

Advancing by eighths

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
1/64	0.04909	0.00019	13/16	8.8357	6.2126	3/8	23.169	42.718
1/32	0.09818	0.00077	7/8	9.0321	6.4918	1/2	23.562	44.179
3/64	0.14726	0.00173	15/16	9.2284	6.7771	5/8	23.955	45.664
1/16	0.19635	0.00307				3/4	24.347	47.173
3/32	0.29452	0.00690	3	9.4248	7.0686	7/8	24.740	48.707
1/8	0.39270	0.01227	1/16	9.6211	7.3662			
5/32	0.49087	0.01917	1/8	9.8175	7.6699	8	25.133	50.265
3/16	0.58905	0.02761	3/16	10.014	7.9798	1/8	25.525	51.849
7/32	0.68722	0.03758	1/4	10.210	8.2958	1/4	25.918	53.456
			5/16	10.407	8.6179	3/8	26.311	55.088
1/4	0.78540	0.04909	3/8	10.603	8.9462	1/2	26.704	56.745
9/32	0.88357	0.06213	7/16	10.799	9.2806	5/8	27.096	58.426
5/16	0.98175	0.07670	1/2	10.996	9.6211	3/4	27.489	60.132
11/32	1.0799	0.09281	9/16	11.192	9.9678	7/8	27.882	61.862
3/8	1.1781	0.11045	5/8	11.388	10.321			
13/32	1.2763	0.12962	11/16	11.585	10.680	9	28.274	63.617
7/16	1.3744	0.15033	3/4	11.781	11.045	1/8	28.667	65.397
15/32	1.4726	0.17257	13/16	11.977	11.416	1/4	29.060	67.201
			7/8	12.174	11.793	3/8	29.452	69.029
1/2	1.5708	0.19635	15/16	12.370	12.177	1/2	29.845	70.882
17/32	1.6690	0.22166				5/8	30.238	72.760
9/16	1.7671	0.24850	4	12.566	12.566	3/4	30.631	74.662
19/32	1.8653	0.27688	1/16	12.763	12.962	7/8	31.023	76.589
5/8	1.9635	0.30680	1/8	12.959	13.364			
21/32	2.0617	0.33824	3/16	13.155	13.772	10	31.416	78.540
11/16	2.1598	0.37122	1/4	13.352	14.186	1/4	32.201	82.516
23/32	2.2580	0.40574	5/16	13.548	14.607	1/2	32.987	86.590
			3/8	13.744	15.033	3/4	33.772	90.763
3/4	2.3562	0.44179	7/16	13.941	15.466			
25/32	2.4544	0.47937	1/2	14.137	15.904	11	34.558	95.033
13/16	2.5525	0.51849	9/16	14.334	16.349	1/4	35.343	99.402
27/32	2.6507	0.55914	5/8	14.530	16.800	1/2	36.128	103.87
7/8	2.7489	0.60132	11/16	14.726	17.257	3/4	36.914	108.43
29/32	2.8471	0.64504	3/4	14.923	17.728			
15/16	2.9452	0.69029	13/16	15.119	18.190	12	37.699	113.10
31/32	3.0434	0.73708	7/8	15.315	18.665	1/4	38.485	117.86
			15/16	15.512	19.147	1/2	39.270	122.72
1	3.1416	0.7854				3/4	40.055	127.68
1/16	3.3379	0.8866	5	15.708	19.635			
1/8	3.5343	0.9940	1/16	15.904	20.129	13	40.841	132.73
3/16	3.7306	1.1075	1/8	16.101	20.629	1/4	41.626	137.89
1/4	3.9270	1.2272	3/16	16.297	21.135	1/2	42.412	143.14
5/16	4.1233	1.3530	1/4	16.493	21.648	3/4	43.197	148.49
3/8	4.3197	1.4849	5/16	16.690	22.166			
7/16	4.5160	1.6230	3/8	16.886	22.691	14	43.982	153.94
1/2	4.7124	1.7671	7/16	17.082	23.221	1/4	44.768	159.48
9/16	4.9087	1.9175	1/2	17.279	23.758	1/2	45.553	165.13
5/8	5.1051	2.0739	9/16	17.475	24.301	3/4	46.338	170.87
11/16	5.3014	2.2365	5/8	17.671	24.850			
3/4	5.4978	2.4053	11/16	17.868	25.406	15	47.124	176.71
13/16	5.6941	2.5802	3/4	18.064	25.967	1/4	47.909	182.65
7/8	5.8905	2.7612	13/16	18.261	26.535	1/2	48.695	188.69
15/16	6.0868	2.9483	7/8	18.457	27.109	3/4	49.480	194.83
			15/16	18.653	27.688			
2	6.2832	3.1416				16	50.265	201.06
1/16	6.4795	3.3410	6	18.850	28.274	1/2	51.836	213.82
1/8	6.6759	3.5466	1/8	19.242	29.465			
3/16	6.8722	3.7583	1/4	19.635	30.680	17	53.407	226.98
1/4	7.0686	3.9761	3/8	20.028	31.919	1/2	54.978	240.53
5/16	7.2649	4.2000	1/2	20.420	33.183			
3/8	7.4613	4.4301	5/8	20.813	34.472	18	56.549	254.47
7/16	7.6576	4.6664	3/4	21.206	35.785	1/2	58.119	268.80
1/2	7.8540	4.9087	7/8	21.598	37.122			
9/16	8.0503	5.1572				19	59.690	283.53
5/8	8.2467	5.4119	7	21.991	38.485	1/2	61.261	298.65
11/16	8.4430	5.6727	1/8	22.384	39.871			
3/4	8.6394	5.9396	1/4	22.776	41.282	20	62.832	314.16

TABLE XXXIX.—CONVERSION FACTORS*

Multiply	by	To obtain
Btu	0.252	Kg-cal
Btu	778	Ft-lb
Btu per min	17.57	Watts
$\frac{\text{Btu}}{\text{sq ft} \times \text{hr}}$	2.71	$\frac{\text{Kg-cal}}{\text{Sq m} \times \text{hr}}$
$\frac{\text{Btu}}{\text{sq ft} \times \text{hr} \times {}^\circ\text{F}}$	4.88	$\frac{\text{Kg-cal}}{\text{Sq m} \times \text{hr} \times {}^\circ\text{C}}$
$\frac{\text{Btu}}{\text{hr} \times {}^\circ\text{F} \times \text{ft}}$	1.49	$\frac{\text{Kg-cal}}{\text{Hr} \times {}^\circ\text{C} \times \text{m}}$
Cm	0.3937	In.
Cm per sec	1.969	Fpm
Cm per sec per sec	0.0328	Fpsps
Cu cm	0.0610	Cu in.
Cu ft	7.48	Gal
Cu ft	0.028	Cu m
Cfm	7.48	Gpm
Cu in.	16.39	Cu cm
Cu in.	0.01639	Liters
Deg (angle)	0.01745	Radians
Ft	30.48	Cm
Fpm	0.508	Cm per sec
Fps	0.682	Mph
Fpsps	30.48	Cm per sec per sec
Ft-lb	0.1383	Kg-m
Ft-lb per sec	1.356	W
Gal	231	Cu in.
Gal	3.785	Liters
Grains	64.80	Mg
G	15.43	Grains
G	0.0353	Oz (avdp)
Hp	42.44	Btu per min
Hp	550	Ft-lb per sec
Hp	746	W
Hp (boiler)	33,480	Btu per hr
Hp-hr	2,545	Btu
Hp-hr	641.7	Kg-cal
Hp-hr	0.746	Kwhr
In.	2.54	Cm
Kg	2.205	Lb
Kg-cal	3.97	Btu
Kg per sq cm	14.22	Lb per sq in.
Km	0.621	Miles
Kw	56.92	Btu/min
Kw	1.341	Hp
Kwhr	3,413	Btu
Kwhr	1.341	Hp-hr
Lb (avdp)	7,000	Grains
Lb (troy)	5,760	Grains
Lb	453.6	Grams
Lb per cu ft	16.0	Kg per cu m
Lb per sq in.	0.0703	Kg per sq cm
Lb per sq in.	0.068	Atm
Liters	0.264	Gal
M	3.821	Ft
Miles	1.609	Km
Mph	44.70	Cm per sec
Mph	88	Fpm
Mph	0.868	Knots
Oz	28.35	G
Oz (troy)	1.097	Oz (avdp)
Oz (fluid)	1.805	Cu in.
Parts per million	0.0584	Grains per gal
Radians	57.3	Deg
Radians per sec	9.55	Rpm
Sq cm	0.155	Sq in.
Sq ft	929.0	Sq cm
Sq m	1.196	Sq yd
Sq miles	640	Acres
Tons	2,000	Lb
Tons (long)	2,240	Lb
Tons (metric)	2,205	Lb
Watts	0.0569	Btu per min
Watts	44.26	Ft-lb per min
Yards	0.9144	M

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* For manometer or pressure-head conversion factors see Table I, p. 15. For viscosity conversion factors see pp. 165 and 489.

TABLE XL.—PROPERTIES OF SOLIDS AND LIQUIDS

For thermal conductivities, densities, and specific heats of solids, see Table XVI, page 202.

For properties of gases, see Table XIX, page 233.

For properties of fuels see pages 486–487.

Metals

Material	Atomic weight	Density, lb per cu ft (room temp)	Specific heat Btu per lb per °F (room temp)	Melting point, °F	Heat of fusion, Btu per lb	Linear coefficient of expansion, per °F × 10,000	Relative electrical conductivity, Cu = 100
Aluminum	26.97	167	0.22	1220	156	0.13	66
Brass		530	0.09			0.10	
Copper	63.57	555	0.095	1982	75	0.09	100
Invar steel (36% Ni)						0.0021	
Iron, gray cast		450	0.13	2400	40	0.06	2.5
Iron, pure	55.84	490	0.11	2795	65	0.065	18
Lead	207.2	710	0.03	621	10	0.16	8
Magnesium	24.32	108	0.24	1203	130	0.145	40
Mercury	200.61	845	0.033	−38	5	1.82*	2
Nickel	58.69	550	0.10	2650	131	0.07	25
Platinum	195.2	1335	0.032	3200	...	0.05	18
Silver	107.9	655	0.056	1762	47	0.107	115
Steel, structural		485	0.11	2550	. .	0.07	12
Tin	118.7	455	0.055	449	25	0.11	16
Tungsten	184.0	1200	0.033	6100	. .	0.02	33
Zinc	65.38	445	0.092	787	46	0.18	31

Liquids

(At atmospheric pressure)

Material	Density, lb per cu ft (room temp)	Specific heat, Btu per lb per °F (room temp)	Boiling point, °F	Latent heat of vaporization, Btu	Freezing point, °F	Viscosity, centipoises, at 70°F
Alcohol, ethyl	49	0.58	173	370	−150	1.1
Ether	45	0.54	95	160	−180	0.25
Gasoline, motor	46	0.50	...	140	.....	0.5
Glycerin	79	0.57	554	. .	.....	700
Kerosene	50	0.50	...	110	.....	2.0
Mercury	845	0.033	675	120	− 38	1.6
Petroleum (machine oil)	53	0.50				
Water	62.3	1.00	212	970	32	1.0

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* Coefficient of cubical expansion.

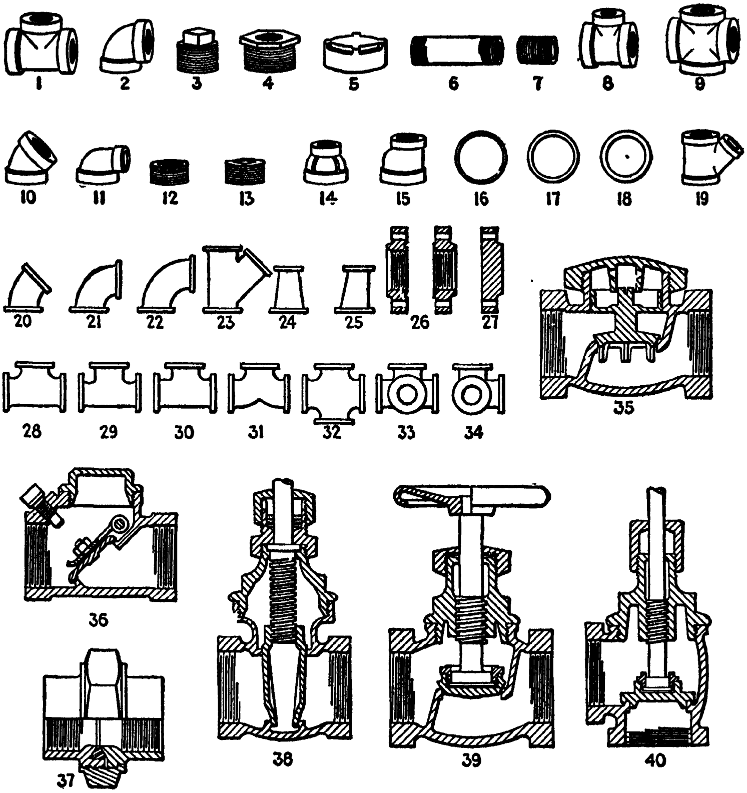


Fig. 193.—Pipe fittings.

Screw Fittings and Pipe:

1. Tee
2. 90-deg elbow
3. Plug
4. Bushing
5. Cap
6. Long nipple
7. Close nipple
8. Reducing tee
9. Cross
10. 45-deg elbow
11. Reducing elbow
12. Faced bushing
13. Countersunk plug
14. Reducing coupling
15. Eccentric reducer
16. Standard pipe
17. Extra-strong pipe
18. Double extra-strong pipe
19. Lateral

Flange Fittings:

20. 45-deg elbow
21. 90-deg elbow
22. Long-radius elbow
23. Lateral
24. Reducer
25. Eccentric reducer
26. Companion flanges or flange union
27. Blind flange
28. Tee
29. Double-sweep tee
30. Single-sweep tee
31. Double-branch elbow
32. Cross
33. Side-outlet tee
34. Side-outlet elbow

Miscellaneous:

35. Horizontal cup or disk check valve
36. Horizontal swing check valve
37. Union (ground metal joint)
38. Typical gate valve, split wedge, inside screw
39. Typical globe valve, renewable seat, inside screw
40. Typical angle valve, renewable seat, inside screw

TABLE XLI.—COPPER TUBE FITTINGS

Adopted by the Refrigeration Valves and Fittings Association and the Society of Automotive Engineers

S.A.E. flare size	Iron pipe size A	Thread size B	C	D	G	I	L	M		O	Q
								Min	Max		
3/16	1/8	3/8 -24	7/16	1/8	3/8	3/8	3/4	0.192	0.197	15/32	15/16
1/4	1/8	7/16 -20	9/16	3/16	3/8	1/2	3/4	0.255	0.260	15/32	15/16
5/16	1/8	1/2 -20	5/8	7/32	3/8	9/16	3/4	0.317	0.322	15/32	15/16
3/8	1/4	5/8 -18	3/4	9/32	7/16	5/8	7/8	0.380	0.385	9/16	1 3/32
7/16	1/4	11/16-16	13/16	5/16	7/16	11/16	1	0.442	0.447	3/4	1 3/16
1/2	3/8	3/4 -16	7/8	13/32	1/2	3/4	1	0.505	0.510	3/4	1 3/16
5/8	1/2	7/8 -14	1	1/2	9/16	7/8	1 1/8	0.630	0.635	15/16	1 7/16
3/4	3/4	1 1/16 -14	1 3/16	11/16	5/8	1	1 3/8	0.755	0.760	1 3/32	1 3/4
1	1	1 3/8 -12	1 3/8	7/8	3/4	1 1/8	1 11/16	1.005	1.010	1 3/8	2

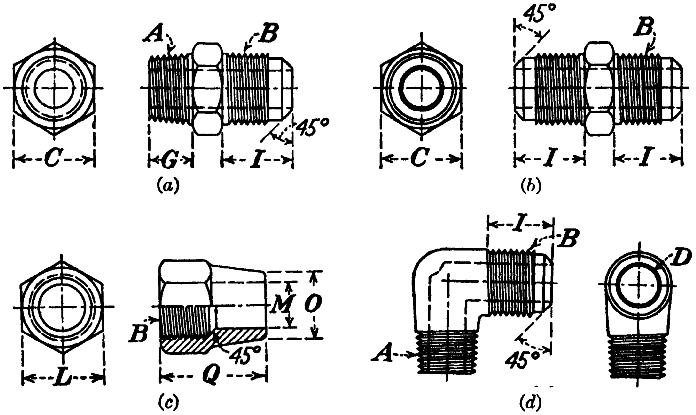


FIG. 194.—Copper tube fittings.

(a) Flare-to-pipe union
(b) Double flare union
(c) Flare nut
(d) Flare-to-pipe elbow

TABLE XLII.—STANDARD DIMENSIONS FOR WROUGHT-IRON AND WROUGHT-STEEL PIPE

Diameter				Circumference		Transverse areas			Length of pipe per sq. ft. of		Length of pipe containing 1 cu. ft., ft.
Nominal internal, in.	Actual external, in.	Actual internal, in.	Thickness, in.	External, in.	Internal, in.	External, sq. in.	Internal, sq. in.	Metal, sq. in.	External, surface, ft.	Internal, surface, ft.	
For standard pipe											
⅛	0 405	0.27	0.068	1 272	0.848	0.129	0.0573	0.0717	9.44	14.15	2,513.0
¼	0.54	0.364	0.088	1.696	1.144	0.229	0.1041	0.1249	7.075	10.49	1,383.3
⅜	0.675	0.494	0.091	2.121	1.552	0.358	0.1917	0.1663	5.657	7.73	751.2
½	0.84	0.623	0.109	2.639	1.957	0.554	0.3048	0.2492	4 547	6.13	472.4
¾	1.05	0.824	0.113	3.299	2.589	0.866	0.5333	0.3327	3.637	4.635	270.0
1	1.315	1.048	0.134	4.131	3.292	1.358	0.8626	0.4954	2.904	3.645	166.9
1¼	1.66	1.38	0.140	5.215	4.335	2 164	1.496	0.668	2.301	2.768	96 25
1½	1.9	1.611	0.145	5.969	5.061	2.835	2.038	0.797	2.01	2.371	70.66
2	2.375	2.067	0.154	7.461	6 494	4 43	3 356	1 074	1.608	1.848	42.91
2½	2.875	2.468	0.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1
3	3.5	3.067	0.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19 5
3½	4	3.548	0.226	12.566	11.146	12.566	9.887	2.679	0.955	1.077	14.57
4	4.5	4.026	0.237	14.137	12.648	15.904	12.73	3.174	0.849	0.949	11.31
5	5.563	5.045	0.259	17.477	15.849	24.306	19.99	4.316	0.687	0.757	7.2
6	6.625	6.065	0.280	20.813	19.054	34.472	28.888	5.584	0.577	0.63	4.98
For extra strength pipe											
⅛	0.405	0.215	0.095	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3,966.392
¼	0.540	0.302	0.119	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2,010.290
⅜	0.675	0.423	0.126	2.121	1.329	0.358	0.141	0 217	5.658	9.030	1,024 689
½	0.840	0.546	0.147	2.639	1.715	0.554	0.234	0.320	4.547	6.995	615.017
¾	1.050	0.742	0.154	3.299	2.331	0.866	0.433	0.433	3.637	5.147	333.016
1	1.315	0.957	0.179	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200 193
1¼	1.660	1.278	0.191	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.256
1½	1.900	1.500	0.200	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487
2	2.375	1.939	0.218	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766
2½	2.875	2.323	0.276	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976
3	3.500	2.900	0.300	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801
3½	4.000	3.364	0.318	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202
4	4.500	3.826	0.337	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525
5	5.563	4.813	0.375	17.477	15.120	24.306	18.194	6.112	0.686	0.793	7.915
6	6.625	5.761	0.432	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524

For bursting pressure in pound per square inch, use Barlow's formula as modified by the A.S.A. Committee on Standardization of Pipe and Tubing:

$$P = \frac{1.75 \times \text{tensile strength}}{\text{outside diameter, in.}} (\text{wall thickness, in.} - 0.10).$$

TABLE XLIII.—THERMAL PROPERTIES OF SATURATED STEAM

For low-pressure table see p. 480.

Pressure, lb. per sq. in. abs.	Temp. deg. fahr.,	Sp. vol., cu. ft. per lb.		Heat content, B.t.u.			Entropy of		
		Steam	Water	Saturated liquid	Evaporation	Saturated vapor,	Water	Evaporation	Saturated steam
p	t	v_g	v_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0.0887 lb.	32	3301.0	0.01602	0.00	1073.4	1073.4	0.0000	2.1834	2.1834
0.5 in. Hg	58.83	1256.9	0.01603	26.88	1058.8	1085.7	0.0533	2.0422	2.0955
1.0 in. Hg	79.06	652.7	0.01607	47.06	1047.8	1094.9	0.0914	1.9451	2.0365
0.5 lb.	79.60	641.8	0.01607	47.60	1047.5	1095.1	0.0925	1.9426	2.0351
2.0 in. Hg	101.17	339.5	0.01613	69.10	1035.7	1104.8	0.1316	1.8468	1.9784
1 lb.	101.76	333.9	0.01614	69.69	1035.3	1105.0	0.1326	1.8442	1.9769
2	126.10	173.96	0.01623	93.97	1021.6	1115.6	0.1750	1.7442	1.9192
3	141.49	118.86	0.01630	109.33	1012.7	1122.0	0.2009	1.6847	1.8856
4	152.99	90.74	0.01636	120.83	1005.9	1126.8	0.2198	1.6420	1.8618
5	162.25	73.61	0.01641	130.10	1000.4	1130.6	0.2348	1.6088	1.8435
6	170.07	62.05	0.01645	137.92	995.8	1133.7	0.2473	1.5814	1.8287
7	176.85	53.70	0.01649	144.71	991.7	1136.4	0.2580	1.5582	1.8162
8	182.87	47.39	0.01652	150.75	988.1	1138.9	0.2674	1.5379	1.8053
9	188.28	42.44	0.01656	156.19	984.8	1141.0	0.2758	1.5200	1.7958
10	193.21	38.45	0.01658	161.13	981.8	1143.0	0.2834	1.5040	1.7874
11	197.75	35.17	0.01661	165.68	979.1	1144.8	0.2903	1.4894	1.7797
12	201.96	32.42	0.01664	169.91	976.5	1146.4	0.2968	1.4760	1.7727
13	205.88	30.08	0.01666	173.85	974.1	1147.9	0.3027	1.4636	1.7663
14	209.56	28.06	0.01669	177.55	971.8	1149.3	0.3082	1.4521	1.7604
14.696	212.00	26.82	0.01670	180.00	970.2	1150.2	0.3119	1.4446	1.7564
15	213.03	26.31	0.01671	181.04	969.6	1150.6	0.3134	1.4414	1.7548
16	216.32	24.76	0.01673	184.35	967.4	1151.8	0.3184	1.4312	1.7496
17	219.43	23.40	0.01676	187.48	965.4	1152.9	0.3230	1.4218	1.7448
18	222.40	22.18	0.01678	190.48	963.5	1154.0	0.3274	1.4127	1.7402
19	225.23	21.08	0.01680	193.34	961.7	1155.0	0.3316	1.4042	1.7358
20	227.96	20.10	0.01682	196.09	959.9	1156.0	0.3356	1.3960	1.7317
21	230.56	19.20	0.01684	198.72	958.2	1156.9	0.3395	1.3883	1.7278
22	233.07	18.38	0.01685	201.25	956.6	1157.8	0.3431	1.3809	1.7240
23	235.49	17.63	0.01687	203.70	955.0	1158.6	0.3466	1.3738	1.7204
24	237.82	16.94	0.01689	206.05	953.4	1159.5	0.3500	1.3670	1.7170
25	240.07	16.31	0.01690	208.33	951.9	1160.2	0.3533	1.3604	1.7137
26	242.25	15.72	0.01692	210.54	950.4	1161.0	0.3564	1.3542	1.7106
27	244.36	15.17	0.01694	212.67	949.0	1161.7	0.3594	1.3481	1.7075
28	246.41	14.66	0.01695	214.75	947.7	1162.4	0.3624	1.3422	1.7046
29	248.40	14.19	0.01697	216.77	946.3	1163.1	0.3652	1.3365	1.7018
30	250.34	13.75	0.01698	218.73	945.0	1163.7	0.3680	1.3310	1.6990
31	252.22	13.33	0.01700	220.64	943.7	1164.4	0.3707	1.3257	1.6964
32	254.05	12.94	0.01701	222.50	942.5	1165.0	0.3732	1.3206	1.6938
33	255.84	12.57	0.01703	224.32	941.2	1165.6	0.3758	1.3156	1.6914
34	257.58	12.23	0.01704	226.09	940.0	1166.1	0.3783	1.3107	1.6890
35	259.28	11.90	0.01706	227.82	938.9	1166.7	0.3807	1.3060	1.6866
36	260.94	11.59	0.01707	229.51	937.7	1167.2	0.3830	1.3014	1.6844
37	262.57	11.29	0.01708	231.17	936.6	1167.8	0.3853	1.2968	1.6822
38	264.16	11.02	0.01710	232.79	935.5	1168.3	0.3876	1.2925	1.6800
39	265.72	10.75	0.01711	234.38	934.4	1168.8	0.3898	1.2882	1.6780
40	267.24	10.50	0.01712	235.93	933.3	1169.2	0.3919	1.2840	1.6759
41	268.74	10.26	0.01713	237.45	932.3	1169.7	0.3940	1.2799	1.6739
42	270.21	10.03	0.01715	238.95	931.2	1170.2	0.3961	1.2759	1.6720
43	271.65	9.808	0.01716	240.42	930.2	1170.6	0.3981	1.2720	1.6701
44	273.06	9.599	0.01717	241.86	929.2	1171.1	0.4000	1.2682	1.6683
45	274.45	9.399	0.01718	243.28	928.2	1171.5	0.4020	1.2645	1.6665
46	275.81	9.207	0.01719	244.67	927.2	1171.9	0.4039	1.2608	1.6647
47	277.14	9.023	0.01720	246.03	926.3	1172.3	0.4057	1.2572	1.6630
48	278.45	8.846	0.01722	247.37	925.4	1172.7	0.4076	1.2537	1.6613
49	279.74	8.676	0.01723	248.68	924.4	1173.1	0.4093	1.2503	1.6596

From "Steam Tables" by J. H. Keenan, by permission of the publisher, The American Society of Mechanical Engineers

TABLE XLIII.—THERMAL PROPERTIES OF SATURATED STEAM.—(*Continued*)

Pressure, lb. per sq. in. abs.	Temp. deg. fahr.,	Sp. vol., cu. ft. per lb.		Heat content, B.t.u.			Entropy of		
		Steam	Water	Saturated liquid	Evaporation	Saturated vapor,	Water	Evaporation	Saturated steam
p	t	v_g	v_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
50	281.01	8.514	0.01724	249.98	923.5	1173.5	0.4111	1.2469	1.6580
51	282.26	8.357	0.01725	251.26	922.6	1173.9	0.4128	1.2436	1.6564
52	283.49	8.206	0.01726	252.52	921.7	1174.3	0.4145	1.2404	1.6549
53	284.70	8.060	0.01727	253.76	920.9	1174.6	0.4162	1.2372	1.6533
54	285.90	7.919	0.01728	254.99	920.0	1175.0	0.4178	1.2340	1.6518
55	287.07	7.783	0.01729	256.19	919.1	1175.3	0.4194	1.2309	1.6504
56	288.23	7.653	0.01730	257.38	918.3	1175.7	0.4210	1.2279	1.6489
57	289.37	7.527	0.01731	258.55	917.5	1176.0	0.4226	1.2249	1.6475
58	290.50	7.405	0.01732	259.71	916.6	1176.4	0.4241	1.2220	1.6461
59	291.62	7.287	0.01734	260.86	915.8	1176.7	0.4256	1.2191	1.6447
60	292.71	7.172	0.01735	261.98	915.0	1177.0	0.4271	1.2162	1.6434
61	293.79	7.062	0.01736	263.09	914.2	1177.3	0.4286	1.2134	1.6420
62	294.85	6.955	0.01737	264.18	913.4	1177.6	0.4300	1.2107	1.6407
63	295.91	6.850	0.01738	265.27	912.7	1177.9	0.4315	1.2080	1.6394
64	296.94	6.749	0.01739	266.33	911.9	1178.2	0.4329	1.2053	1.6382
65	297.97	6.652	0.01740	267.39	911.1	1178.5	0.4343	1.2026	1.6369
66	298.98	6.556	0.01741	268.43	910.4	1178.8	0.4356	1.2001	1.6357
67	299.99	6.464	0.01742	269.47	909.6	1179.1	0.4370	1.1975	1.6345
68	300.98	6.375	0.01743	270.49	908.9	1179.4	0.4384	1.1950	1.6333
69	301.96	6.288	0.01744	271.50	908.2	1179.6	0.4397	1.1924	1.6321
70	302.92	6.203	0.01744	272.49	907.4	1179.9	0.4410	1.1900	1.6310
71	303.88	6.121	0.01745	273.48	906.7	1180.2	0.4423	1.1876	1.6298
72	304.82	6.041	0.01746	274.45	906.0	1180.5	0.4435	1.1852	1.6287
73	305.76	5.963	0.01747	275.42	905.3	1180.7	0.4448	1.1828	1.6276
74	306.68	5.887	0.01748	276.37	904.6	1181.0	0.4460	1.1805	1.6265
75	307.60	5.813	0.01749	277.32	903.9	1181.2	0.4473	1.1782	1.6254
76	308.50	5.741	0.01750	278.25	903.2	1181.5	0.4485	1.1759	1.6244
77	309.39	5.671	0.01751	279.17	902.6	1181.7	0.4497	1.1736	1.6233
78	310.28	5.602	0.01752	280.09	901.9	1182.0	0.4509	1.1714	1.6223
79	311.16	5.535	0.01753	281.00	901.2	1182.2	0.4520	1.1692	1.6212
80	312.03	5.470	0.01754	281.90	900.5	1182.4	0.4532	1.1670	1.6202
81	312.88	5.406	0.01755	282.79	899.9	1182.7	0.4544	1.1649	1.6192
82	313.74	5.343	0.01756	283.67	899.2	1182.9	0.4555	1.1627	1.6182
83	314.58	5.282	0.01756	284.55	898.6	1183.1	0.4566	1.1606	1.6173
84	315.42	5.222	0.01757	285.42	897.9	1183.4	0.4578	1.1586	1.6163
85	316.25	5.164	0.01758	286.28	897.3	1183.6	0.4589	1.1565	1.6154
86	317.06	5.107	0.01759	287.13	896.7	1183.8	0.4599	1.1545	1.6144
87	317.88	5.051	0.01760	287.97	896.0	1184.0	0.4610	1.1524	1.6135
88	318.68	4.997	0.01761	288.80	895.4	1184.2	0.4621	1.1505	1.6126
89	319.48	4.944	0.01762	289.63	894.8	1184.4	0.4632	1.1485	1.6116
90	320.27	4.892	0.01763	290.45	894.2	1184.6	0.4642	1.1465	1.6107
91	321.05	4.841	0.01764	291.26	893.6	1184.8	0.4652	1.1446	1.6099
92	321.83	4.791	0.01764	292.07	893.0	1185.0	0.4663	1.1427	1.6090
93	322.60	4.742	0.01765	292.87	892.4	1185.2	0.4673	1.1408	1.6081
94	323.37	4.694	0.01766	293.67	891.8	1185.4	0.4683	1.1389	1.6072
95	324.13	4.647	0.01767	294.47	891.2	1185.6	0.4693	1.1370	1.6064
96	324.88	4.602	0.01768	295.25	890.6	1185.8	0.4703	1.1352	1.6055
97	325.62	4.557	0.01768	296.02	890.0	1186.0	0.4713	1.1334	1.6047
98	326.37	4.512	0.01769	296.80	889.4	1186.2	0.4723	1.1316	1.6038
99	327.10	4.468	0.01770	297.57	888.8	1186.4	0.4732	1.1298	1.6030
100	327.83	4.426	0.01771	298.33	888.2	1186.6	0.4742	1.1280	1.6022
102	329.27	4.344	0.01773	299.83	887.1	1186.9	0.4761	1.1245	1.6006
104	330.68	4.265	0.01774	301.30	886.0	1187.3	0.4779	1.1211	1.5990
106	332.08	4.189	0.01776	302.76	884.9	1187.6	0.4798	1.1177	1.5974
108	333.44	4.115	0.01777	304.19	883.8	1188.0	0.4816	1.1144	1.5959

TABLE XLIII.—THERMAL PROPERTIES OF SATURATED STEAM.—(*Continued*)

Pressure, lb. per sq. in. abs.	Temp. deg. fahr.,	Sp. vol., cu. ft. per lb.		Heat content, B.t.u.			Entropy of		
		Steam	Water	Saturated liquid	Evaporation	Saturated vapor,	Water	Evaporation	Saturated steam
p	t	v_g	v_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
110	334.79	4.044	0.01779	305.61	882.7	1188.3	0.4834	1.1111	1.5944
112	336.12	3.976	0.01780	307.00	881.6	1188.6	0.4851	1.1079	1.5930
114	337.43	3.910	0.01782	308.36	880.6	1188.9	0.4868	1.1048	1.5915
116	338.72	3.846	0.01783	309.71	879.5	1189.2	0.4885	1.1017	1.5901
118	340.01	3.784	0.01785	311.05	878.5	1189.5	0.4901	1.0986	1.5887
120	341.26	3.725	0.01786	312.37	877.4	1189.8	0.4918	1.0956	1.5874
122	342.50	3.670	0.01788	313.67	876.4	1190.1	0.4934	1.0926	1.5860
124	343.73	3.615	0.01789	314.96	875.4	1190.4	0.4950	1.0897	1.5847
126	344.94	3.560	0.01791	316.23	874.4	1190.6	0.4965	1.0868	1.5834
128	346.14	3.505	0.01792	317.49	873.4	1190.9	0.4981	1.0840	1.5821
130	347.31	3.451	0.01794	318.73	872.4	1191.2	0.4996	1.0812	1.5808
132	348.48	3.401	0.01795	319.95	871.5	1191.4	0.5011	1.0784	1.5796
134	349.64	3.353	0.01796	321.17	870.5	1191.7	0.5026	1.0757	1.5783
136	350.78	3.306	0.01798	322.37	869.6	1191.9	0.5041	1.0730	1.5771
138	351.91	3.260	0.01799	323.56	868.6	1192.2	0.5056	1.0703	1.5759
140	353.03	3.216	0.01801	324.74	867.7	1192.4	0.5070	1.0677	1.5747
142	354.14	3.173	0.01802	325.91	866.7	1192.6	0.5084	1.0651	1.5735
144	355.22	3.130	0.01804	327.06	865.8	1192.9	0.5098	1.0625	1.5724
146	356.31	3.089	0.01805	328.20	864.9	1193.1	0.5112	1.0600	1.5712
148	357.37	3.049	0.01806	329.32	864.0	1193.3	0.5126	1.0575	1.5701
150	358.43	3.010	0.01808	330.44	863.1	1193.5	0.5140	1.0550	1.5690
152	359.47	2.972	0.01809	331.54	862.2	1193.7	0.5153	1.0526	1.5679
154	360.51	2.935	0.01810	332.64	861.3	1193.9	0.5166	1.0502	1.5668
156	361.53	2.900	0.01812	333.72	860.4	1194.1	0.5180	1.0478	1.5658
158	362.54	2.864	0.01813	334.80	859.5	1194.3	0.5193	1.0454	1.5647
160	363.55	2.830	0.01814	335.86	858.7	1194.5	0.5205	1.0431	1.5636
162	364.54	2.797	0.01816	336.91	857.8	1194.7	0.5218	1.0408	1.5626
164	365.52	2.764	0.01817	337.95	857.0	1194.9	0.5230	1.0385	1.5616
166	366.50	2.733	0.01818	338.99	856.1	1195.1	0.5243	1.0363	1.5606
168	367.46	2.701	0.01819	340.01	855.2	1195.3	0.5255	1.0340	1.5596
170	368.42	2.671	0.01821	341.03	854.4	1195.4	0.5268	1.0318	1.5586
172	369.37	2.641	0.01822	342.04	853.6	1195.6	0.5280	1.0296	1.5576
174	370.31	2.612	0.01823	343.04	852.7	1195.8	0.5292	1.0275	1.5566
176	371.24	2.584	0.01825	344.03	851.9	1196.0	0.5304	1.0253	1.5557
178	372.16	2.556	0.01826	345.01	851.1	1196.1	0.5315	1.0232	1.5548
180	373.08	2.529	0.01827	345.99	850.3	1196.3	0.5327	1.0211	1.5538
182	374.00	2.502	0.01828	346.97	849.5	1196.4	0.5339	1.0190	1.5529
184	374.90	2.476	0.01829	347.94	848.6	1196.6	0.5350	1.0169	1.5520
186	375.78	2.451	0.01831	348.89	847.9	1196.8	0.5362	1.0149	1.5511
188	376.67	2.425	0.01832	349.83	847.1	1196.9	0.5373	1.0129	1.5502
190	377.55	2.401	0.01833	350.77	846.3	1197.0	0.5384	1.0109	1.5493
192	378.42	2.377	0.01834	351.70	845.5	1197.2	0.5395	1.0089	1.5484
194	379.27	2.353	0.01835	352.61	844.7	1197.3	0.5406	1.0070	1.5475
196	380.13	2.330	0.01837	353.53	844.0	1197.5	0.5417	1.0050	1.5467
198	380.97	2.307	0.01838	354.43	843.2	1197.6	0.5427	1.0031	1.5458
200	381.82	2.285	0.01839	355.3	842.4	1197.8	0.5438	1.0012	1.5450
205	383.89	2.231	0.01842	357.6	840.5	1198.1	0.5465	0.9964	1.5429
210	385.93	2.180	0.01844	359.8	838.6	1198.4	0.5491	0.9918	1.5409
215	387.93	2.131	0.01847	361.9	836.8	1198.7	0.5516	0.9873	1.5389
220	389.89	2.084	0.01850	364.0	835.0	1199.0	0.5540	0.9829	1.5369
225	391.81	2.039	0.01853	366.1	833.2	1199.3	0.5565	0.9786	1.5350
230	393.70	1.996	0.01856	368.1	831.4	1199.6	0.5588	0.9743	1.5332
235	395.56	1.955	0.01859	370.2	829.7	1199.8	0.5612	0.9702	1.5313
240	397.40	1.916	0.01861	372.1	827.9	1200.1	0.5635	0.9661	1.5295
245	399.20	1.878	0.01864	374.1	826.2	1200.3	0.5658	0.9620	1.5278

Table XLIII.—Thermal Properties of Saturated Steam.—(*Continued*)

Pressure, lb. per sq. in. abs.	Temp. deg. fahr.,	Sp. vol., cu. ft. per lb.		Heat content, B.t.u.			Entropy of		
		Steam	Water	Saturated liquid	Evaporation	Saturated vapor,	Water	Evaporation	Saturated steam
p	t	v_g	v_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
250	400.97	1.841	0.01867	376.0	824.5	1200.5	0.5680	0.9581	1.5261
260	404.43	1.772	0.01872	379.8	821.2	1201.0	0.5723	0.9504	1.5227
270	407.79	1.708	0.01877	383.4	818.0	1201.4	0.5765	0.9430	1.5194
280	411.06	1.649	0.01882	387.0	814.7	1201.8	0.5805	0.9357	1.5163
290	414.24	1.593	0.01887	390.5	811.6	1202.1	0.5845	0.9287	1.5132
300	417.33	1.541	0.01892	393.9	808.5	1202.4	0.5883	0.9220	1.5102
320	423.29	1.447	0.01901	400.5	802.5	1203.0	0.5957	0.9089	1.5046
340	428.96	1.363	0.01910	406.7	796.6	1203.4	0.6027	0.8965	1.4992
360	434.39	1.288	0.01918	412.8	790.9	1203.7	0.6094	0.8846	1.4940
380	439.59	1.221	0.01927	418.6	785.3	1203.9	0.6157	0.8733	1.4891
400	444.58	1.160	0.0194	424.2	779.8	1204.1	0.6218	0.8625	1.4843
450	456.27	1.030	0.0196	437.4	766.7	1204.1	0.6361	0.8371	1.4732
500	466.99	0.926	0.0198	449.7	754.0	1203.7	0.6493	0.8137	1.4630
550	476.90	0.840	0.0200	461.3	741.7	1203.0	0.6616	0.7920	1.4535
600	486.17	0.768	0.0202	472.3	729.8	1202.1	0.6731	0.7716	1.4447
650	494.84	0.706	0.0204	482.8	718.2	1201.0	0.6840	0.7524	1.4364
700	503.04	0.653	0.0206	492.9	706.8	1199.7	0.6943	0.7342	1.4285
750	510.79	0.606	0.0208	502.5	695.7	1198.3	0.7041	0.7170	1.4210
800	518.18	0.565	0.0209	511.8	684.9	1196.7	0.7135	0.7004	1.4139
850	525.20	0.529	0.0211	520.8	674.2	1195.0	0.7225	0.6848	1.4071
900	531.95	0.497	0.0213	529.5	663.8	1193.3	0.7311	0.6694	1.4005
950	538.40	0.468	0.0215	537.9	653.6	1191.5	0.7394	0.6549	1.3942
1000	544.58	0.442	0.0217	546.0	643.5	1189.6	0.7473	0.6408	1.3881
1050	550.53	0.418	0.0219	554.0	633.6	1187.6	0.7550	0.6273	1.3822
1100	556.28	0.396	0.0222	561.7	623.9	1185.6	0.7624	0.6141	1.3765
1150	561.81	0.376	0.0224	569.2	614.3	1183.5	0.7695	0.6014	1.3709
1200	567.14	0.358	0.0226	576.5	604.9	1181.4	0.7764	0.5891	1.3656
1250	572.30	0.342	0.0228	583.6	595.6	1179.2	0.7831	0.5772	1.3603
1300	577.32	0.326	0.0230	590.6	586.3	1177.0	0.7897	0.5654	1.3552
1350	582.21	0.312	0.0232	597.5	577.2	1174.7	0.7962	0.5540	1.3501
1400	586.96	0.298	0.0235	604.3	568.1	1172.4	0.8024	0.5428	1.3452
1450	591.58	0.286	0.0237	611.0	559.1	1170.0	0.8086	0.5318	1.3404
1500	596.08	0.274	0.0239	617.5	550.2	1167.6	0.8146	0.5212	1.3357
1600	604.74	0.253	0.0244	630.2	532.6	1162.7	0.8262	0.5003	1.3265
1700	612.98	0.234	0.0249	642.5	515.0	1157.5	0.8373	0.4801	1.3174
1800	620.86	0.217	0.0254	654.7	497.2	1151.8	0.8482	0.4601	1.3083
1900	628.30	0.201	0.0260	666.8	478.9	1145.7	0.8589	0.4402	1.2991
2000	635.61	0.188	0.0265	679.0	460.0	1139.0	0.8696	0.4200	1.2896
2200	649.25	0.162	0.0277	703.7	420.0	1123.8	0.8912	0.3788	1.2700
2400	661.94	0.140	0.0292	729.4	376.4	1105.8	0.9133	0.3356	1.2489
2600	673.82	0.121	0.0310	756.7	327.8	1084.5	0.9364	0.2892	1.2256
2800	684.91	0.102	0.0333	786.7	272.3	1058.9	0.9618	0.2379	1.1997
3000	695.25	0.084	0.0367	823.1	202.5	1025.6	0.9922	0.1754	1.1676
3200	704.91	0.060	0.0459	887.0	75.9	962.9	1.0461	0.0651	1.1112
3226	706.10	0.052	0.0522	925.0	00.0	925.0	1.0785	0.0000	1.0785

TABLE XLIV.—PROPERTIES OF SUPERHEATED STEAM

Abs. press. lb. per sq. in. (sat. temp.)		Temperature—deg. fahr.								
		200	250	300	350	400	450	500	600	700
1	*v*	392.5	422 3	452.1	481 9	511 7	541 6	571 3	630 9	690 6
	h	1149 8	1172 4	1195 0	1217 8	1240 7	1263 9	1287 2	1334 6	1383 0
(101.76)	*s*	2.0503	2.0835	2.1142	2 1433	2 1707	2 1968	2 2218	2 2688	2 3125
5	*v*	78 17	84.19	90 21	96 21	102 19	108 19	114 16	126 11	138 05
	h	1148 2	1171 3	1194 2	1217 2	1240 2	1263 5	1286 9	1334 4	1382 9
(162 25)	*s*	1.8712	1.9050	1.9361	1.9654	1 9930	2.0192	2 0443	2 0914	2.1351
10	*v*	38 88	41 92	44 98	48 00	51 01	54.02	57 02	63 01	68 99
	h	1146 3	1169 9	1193 2	1216 4	1239 6	1262 9	1286 4	1334 0	1382 6
(193.21)	*s*	1 7925	1.8271	1.8587	1.8838	1.9161	1.9430	1.9676	2 0148	2 0586
14.696	*v*		28 40	30 52	32.59	34 65	36 70	38 75	42 83	46 91
	h		1168 6	1192 2	1215 6	1239 0	1262 4	1286 0	1333 7	1382 4
(212.00)	*s*		1.7832	1.8154	1.8452	1.8731	1.8996	1.9249	1 9722	2.0161
20	*v*		20 79	22 36	23 90	25.43	26 94	28 45	31.46	34.46
	h		1167 0	1191 1	1214 8	1238 3	1261 9	1285 5	1333.4	1382 1
(227.96)	*s*		1 7474	1.7802	1.8104	1 8386	1.8652	1 8906	1.9380	1.9819
40	*v*			11 044	11 843	12.623	13 392	14.161	15 682	17 190
	h			1186 5	1211 3	1235 6	1259 7	1283 7	1332 0	1381 2
(267.24)	*s*			1.6990	1.7309	1.7599	1.7871	1 8128	1 8607	1.9050
60	*v*			7 260	7.821	8.353	8 878	9 398	10 423	11 435
	h			1181 2	1207 7	1232 8	1257 4	1281 9	1330.7	1380 2
(292.71)	*s*			1.6488	1.6827	1 7128	1.7406	1 7667	1 8151	1 8597
80	*v*				5 806	6 217	6 618	7 015	7 793	8 558
	h				1203 8	1229 9	1255 2	1280 0	1329.3	1379 2
(312.03)	*s*				1.6473	1 6785	1.7070	1.7336	1.7824	1.8274
100	*v*				4 594	4 934	5 263	5 585	6 215	6 831
	h				1199 7	1226 9	1252 8	1278 0	1327.9	1378 2
(327.83)	*s*				1 6185	1 6512	1.6805	1.7075	1.7569	1 8023
120	*v*				3 781	4 077	4 359	4 632	5 162	5.680
	h				1195 2	1223 8	1250 4	1276 1	1326 5	1377 2
(341.26)	*s*				1 5940	1.6283	1.6584	1.6859	1 7359	1.7816
140	*v*					3 465	3 713	3 951	4 410	4 857
	h					1220 5	1247 9	1274 1	1325.1	1376 2
(353 03)	*s*					1.6084	1 6393	1.6674	1 7179	1 7640
160	*v*					3 005	3 227	3 440	3 846	4.240
	h					1217 1	1245 4	1272 1	1323 7	1375 2
(363 55)	*s*					1.5906	1.6224	1.6510	1.7022	1.7487
180	*v*					2 646	2 849	3 041	3 407	3.760
	h					1213 5	1242 7	1270 0	1322 2	1374 2
(373.08)	*s*					1 5742	1.6073	1.6364	1.6882	1.7351
200	*v*					2 358	2.547	2.722	3 056	3· 376
	h					1209 8	1240.0	1267 9	1320 8	1373 1
(381.82)	*s*					1.5592	1.5934	1.6231	1.6756	1.7228
220	*v*					2 122	2.299	2 462	2 769	3 062
	h					1205 9	1237 3	1265 7	1319.3	1372.1
(389.89)	*s*					1 5450	1.5805	1.6109	1 6641	1.7117
240	*v*					1 9250	2 092	2 244	2 529	2.800
	h					1201 9	1234 4	1263 5	1317 8	1371 0
(397.40)	*s*					1.5317	1.5684	1 5996	1.6534	1.7014

TABLE XLIV.—PROPERTIES OF SUPERHEATED STEAM.—(*Continued*)

Abs. press. lb. per sq. in. (sat. temp.)		Temperature—deg. fahr.								
		450	500	550	600	650	700	800	900	1000
260 (404.43)	*v*	1 9165	2 060	2.195	2 327	2.454	2.579	2.823	3.062	3 295
	h	1231 5	1261.3	1289 3	1316.3	1343 1	1370.0	1423.3	1476 3	1529 7
	s	1.5571	1.5890	1 6174	1.6435	1.6682	1.6919	1.7360	1.7765	1.8144
280 (411.06)	*v*	1 7655	1 9019	2 030	2.153	2 272	2.389	2 617	2.839	3 056
	h	1228.4	1259 0	1287 5	1314.8	1341 9	1368 9	1422 5	1475.7	1529 3
	s	1 5462	1 5790	1.6078	1.6343	1.6592	1.6830	1.7274	1.7680	1.8060
300 (417.33)	*v*	1 6347	1 7648	1.8863	2.002	2 115	2.224	2 438	2 646	2.849
	h	1225 3	1256 7	1285 6	1313 3	1340.6	1367 8	1421 7	1475 1	1528 9
	s	1.5359	1 5695	1.5988	1.6256	1.6508	1.6747	1.7193	1.7601	1.7983
350 (431.71)	*v*	1 3712	1 4899	1.5981	1.7003	1.7989	1.8945	2 080	2 260	2.435
	h	1217.1	1250 7	1280 9	1309 4	1337.3	1365 1	1419.8	1473.6	1527.9
	s	1.5117	1.5477	1.5783	1.6059	1.6316	1.6561	1.7012	1.7424	1.7809
400 (444.58)	*v*	1.1726	1.2828	1 3817	1.4740	1.5618	1 6472	1.8119	1.9704	2.125
	h	1208 3	1244 3	1276 0	1305.5	1334.0	1362 3	1417.7	1472 1	1526 8
	s	1.4892	1.5276	1 5599	1.5884	1.6146	1.6396	1.6854	1.7270	1.7658
450 (456.27)	*v*	......	1.1204	1.2127	1.2972	1.377	1 455	1.603	1.746	1.883
	h	......	1237 6	1271.0	1301.5	1330 7	1359.4	1415.7	1470 6	1525.8
	s	......	1.5091	1.5430	1.5725	1.5994	1.6248	1.6714	1.7133	1.7524
500 (466.99)	*v*	......	0 9905	1 078	1.156	1.230	1.301	1.437	1.566	1 690
	h	......	1230 5	1265.8	1297 3	1327 3	1356 6	1413.6	1469.1	1524 8
	s	......	1.4915	1.5274	1.5579	1.5855	1.6113	1.6586	1.7009	1 7404
550 (476.90)	*v*	......	0 8823	0.9658	1 040	1.109	1 175	1 300	1.418	1.532
	h	......	1222.8	1260.3	1293.1	1323 8	1353.6	1411.5	1467.6	1523.8
	s	.	1.4747	1.5126	1.5443	1.5726	1.5990	1.6468	1.6897	1 7296
600 (486.17)	*v*	....	0.7922	0.8728	0.9431	1.008	1 069	1.186	1 295	1 400
	h	.....	1214.7	1254.6	1288.7	1320 2	1350.6	1409 3	1466.1	1522.8
	s	.. .	1.4582	1.4986	1.5316	1.5607	1.5874	1.6360	1.6794	1.7196
700 (503.04)	*v*	......	.....	0.7251	0 7905	0.8490	0 9038	1 006	1.103	1.193
	h	......	......	1242.4	1279 7	1312.9	1344.5	1404.9	1463.0	1520 8
	s	......	....	1.4720	1.5080	1 5387	1.5665	1.6165	1.6608	1.7018
800 (518.18)	*v*	......	......	0 6128	0 6750	0.7293	0.7791	0.8723	0.9577	1 037
	h	.....	....	1228 8	1270 1	1305 4	1338 2	1400 4	1459 9	1518 9
	s	......	.....	1.4462	1.4862	1.5187	1.5477	1.5992	1.6446	1.6864
1000 (544.58)	*v*	......	......	0 4495	0.5111	0.5602	0.6040	0 6837	0 7547	0.8199
	h	......	......	1196 5	1248 7	1289.2	1324 9	1391.0	1453.6	1514.8
	s	......	......	1 3949	1.4455	1.4829	1.5144	1.5691	1.6169	1.6603
1200 (567.14)	*v*	......	.....	......	0.3985	0.4462	0.4865	0.5578	0.6195	0.6750
	h	......	......	......	1223.4	1271 3	1310.6	1381 1	1447.2	1510.8
	s	......	......	......	1.4058	1.4501	1.4848	1.5431	1.5937	1.6388
1600 (604.74)	*v*	......	......	.....	......	0.2999	0.3384	0.4002	0.4503	0.4935
	h	......	......	......	......	1228.3	1278.6	1360.0	1434.3	1502.9
	s	......	......	......	......	1.3870	1.4313	1.4986	1.5555	1.6042
2000 (635.61)	*v*	......	......	......	......	0.2044	0.2468	0.3047	0.3486	0.3847
	h	......	....	......	......	1168.9	1240.7	1337.0	1421.1	1494.7
	s	......	......	......	......	1.3168	1.3802	1.4599	1.5242	1.5765
3000 (695.3)	*v*	......	......	......	......	......	0.0983	0.1742	0.2118	0.2396
	h	......	......	......	......	......	1066.3	1271.1	1384.3	1472.9
	s	......	......	......	......	......	1.2028	1.3737	1.4602	1.5233

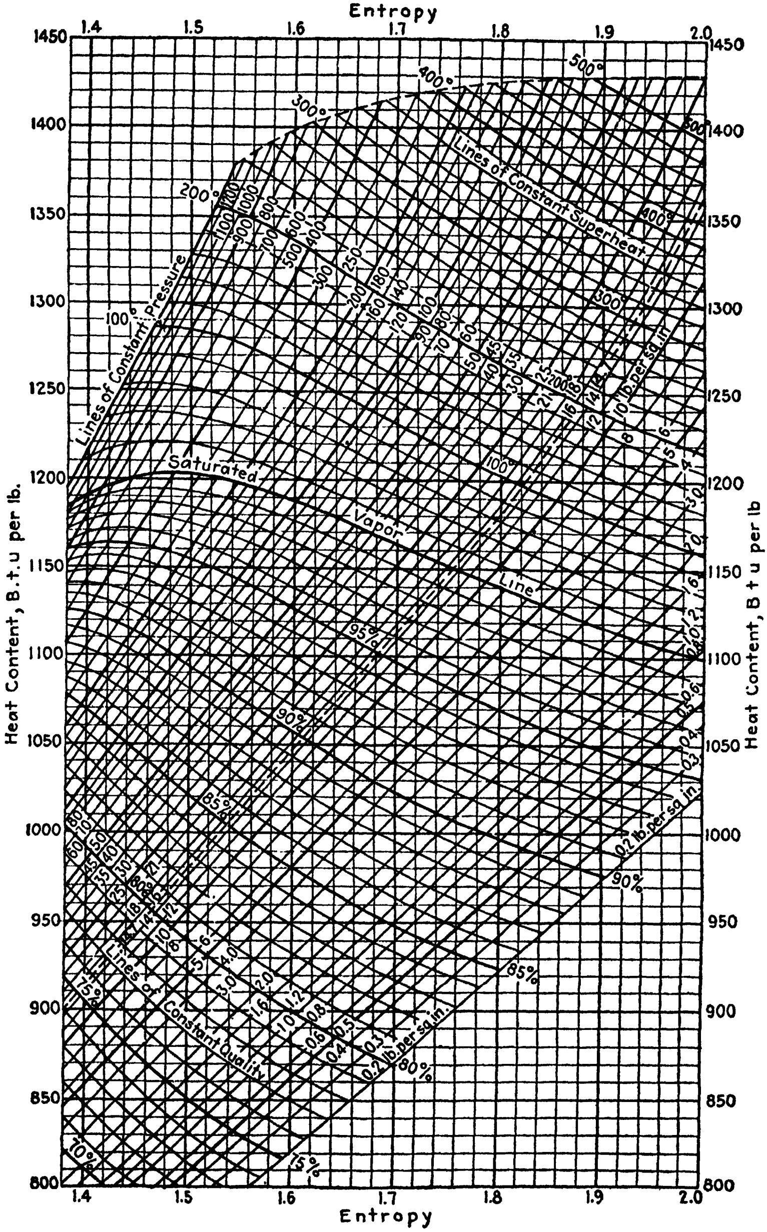


FIG. 195.—Heat-content-entropy or Mollier diagram for steam. (Keenan-A.S.M.E. values.)

TABLE XLV.—PROPERTIES OF LOW-PRESSURE STEAM AND STEAM-AIR MIXTURES

(For psychrometric chart see p. 248.)

Temperature, °F	Steam only				Saturated mixture		
	Pressure saturated steam		Volume saturated steam, cu ft per lb	Enthalpy saturated steam, Btu per lb	Wt of vapor per lb dry air, grains	Volume saturated mixture, cu ft per lb	Enthalpy saturated mixture, Btu per lb
	Lb per sq in.	In. Hg					
40	0.1217	0.2478	2,444	1079.3	36.36	12.69	15.19
41	0.1265	0.2576	2,356	1079.7	37.80	12.72	15.66
42	0.1315	0.2677	2,271	1080.2	39.31	12.75	16.13
43	0.1367	0.2783	2,190	1080.6	40.88	12.78	16.62
44	0.1420	0.2891	2,112	1081.0	42.48	12.81	17.11
45	0.1475	0.3003	2,036	1081.5	44.14	12.84	17.61
46	0.1532	0.3119	1,964	1081.9	45.87	12.87	18.12
47	0.1591	0.3239	1,895	1082.4	47.66	12.90	18.64
48	0.1652	0.3364	1,829	1082.8	49.50	12.93	19.16
49	0.1715	0.3492	1,765	1083.2	51.42	12.96	19.70
50	0.1780	0.3624	1,703	1083.7	53.38	12.99	20.25
51	0.1848	0.3763	1,644	1084.1	55.45	13.02	20.80
52	0.1918	0.3905	1,588	1084.5	57.58	13.06	21.38
53	0.1989	0.4050	1,533	1085.0	59.74	13.09	21.95
54	0.2063	0.4200	1,481	1085.4	61.99	13.12	22.55
55	0.2140	0.4357	1,431	1085.8	64.34	13.15	23.15
56	0.2219	0.4518	1,382	1086.3	66.75	13.19	23.77
57	0.2300	0.4685	1,336	1086.7	69.23	13.22	24.40
58	0.2384	0.4854	1,291	1087.1	71.82	13.25	25.05
59	0.2471	0.5031	1,248	1087.6	74.48	13.29	25.70
60	0.2561	0.5214	1,207	1088.0	77.21	13.32	26.37
61	0.2654	0.5404	1,167	1088.4	80.08	13.35	27.06
62	0.2749	0.5597	1,128	1088.9	83.02	13.39	27.76
63	0.2848	0.5799	1,091	1089.3	86.03	13.42	28.48
64	0.2949	0.6004	1,056	1089.7	89.18	13.46	29.21
65	0.3054	0.6218	1,021	1090.2	92.40	13.49	29.96
66	0.3162	0.6438	988.4	1090.6	95.76	13.53	30.73
67	0.3273	0.6664	956.6	1091.0	99.19	13.57	31.51
68	0.3388	0.6898	925.9	1091.5	102.8	13.60	32.31
69	0.3506	0.7138	896.3	1091.9	106.4	13.64	33.12
70	0.3628	0.7387	867.9	1092.3	110.2	13.68	33.96
71	0.3754	0.7643	840.4	1092.8	114.2	13.71	34.83
72	0.3883	0.7906	813.9	1093.2	118.2	13.75	35.70
73	0.4016	0.8177	788.4	1093.6	122.4	13.79	36.60
74	0.4153	0.8456	763.8	1094.1	126.6	13.83	37.51
75	0.4295	0.8745	740.0	1094.5	131.1	13.87	38.46
76	0.4440	0.9040	717.1	1094.9	135.7	13.91	39.42
77	0.4590	0.9345	694.9	1095.4	140.4	13.95	40.40
78	0.4744	0.9659	673.6	1095.8	145.3	13.99	41.42
79	0.4903	0.9983	653.0	1096.2	150.3	14.03	42.46
80	0.5067	1.032	633.1	1096.6	155.5	14.08	43.51
81	0.5236	1.066	613.9	1097.1	160.9	14.12	44.61
82	0.5409	1.101	595.3	1097.5	166.4	14.16	45.72
83	0.5588	1.138	577.4	1097.9	172.1	14.21	46.88
84	0.5772	1.175	560.2	1098.4	178.0	14.26	48.05
85	0.5960	1.214	543.5	1098.8	184.0	14.30	49.24
86	0.6153	1.253	527.3	1099.2	190.3	14.34	50.47
87	0.6352	1.293	511.7	1099.7	196.7	14.39	51.74
88	0.6555	1.335	496.7	1100.1	203.3	14.44	53.02
89	0.6765	1.377	482.1	1100.5	210.1	14.48	54.35
90	0.6980	1.421	468.0	1100.9	217.1	14.53	55.70

THE COMFORT CHART

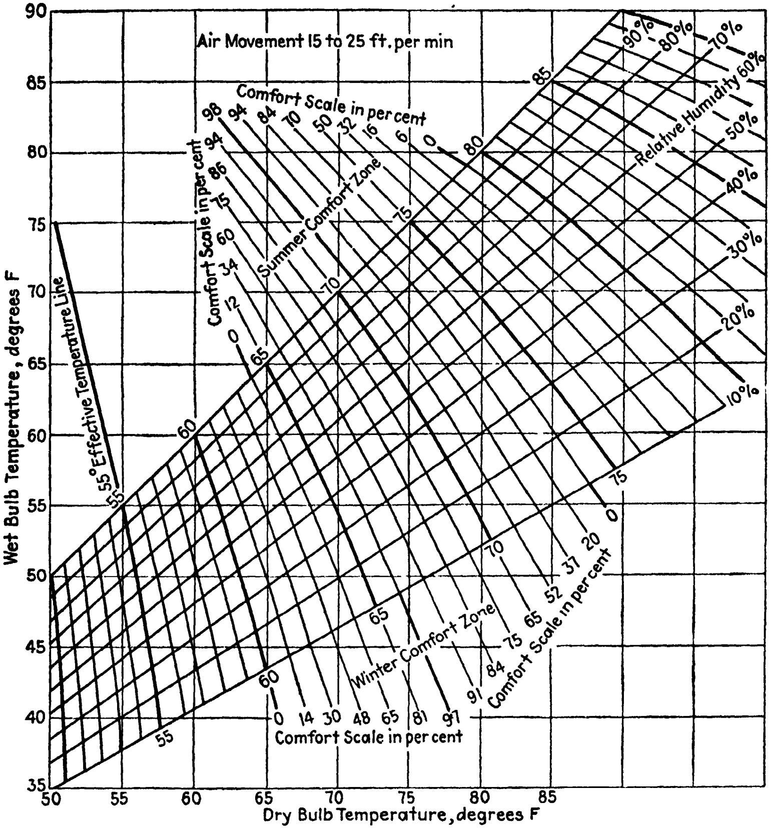


FIG. 196.—Comfort chart developed by A.S.H.V.E. Research Laboratory. (*Used with permission.*) See *Trans.* A.S.H.V.E., Vol. 38, 1932, "How to Use the Effective Temperature Index and Comfort Charts."

Notes on application of comfort chart:

This chart applies to persons normally clothed and slightly active.

While the "effective temperature lines" are essentially lines of *equal comfort*, the zone of maximum comfort is usually indicated as between 30 and 70 per cent relative humidity.

Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices, and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and other spaces where the exposure is less than 3 hr.

PLOTTING CURVES WITH THE USE OF LOGARITHMIC PAPER

Equations representing the relationship between two quantities are chiefly in the form of $Q = Kh^{\pm n} \pm Z$.

When the expression involves the first power, or $n = 1$, the equation takes the form of $Q = Kh \pm Z$ and will plot as a straight line on regular cross-section paper.

When n has some value other than 1, the line expressing the Q-K relationship will be a curve, when plotted on plain paper. If the data be plotted on logarithmic paper, the line will be straight when $Z = 0$.

EQUATIONS OF THE FORM $Q = Kh^n$

Take the equation of form

$$Q = Kh^n, \text{ or } \log Q = n \log h + \log K.$$

In this equation n is the slope of the line or the tangent of the angle which the line makes with the h-axis, and K is the intercept on the Q-axis.

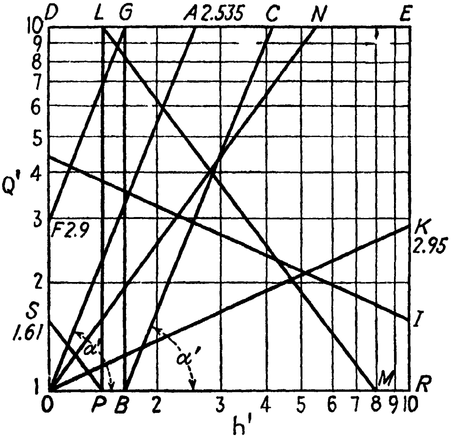


FIG. 197.

In Fig. 197, let BC be the line plotted on logarithmic paper for the flow of water over a weir. Draw AO parallel to BC and passing through the origin O.

The equation of the line BC is $\log Q = n \log h + \log K$; when this line is shifted to AO, K, the intercept on the Q-axis, becomes 1, and $\log K = 0$, therefore, the equation of AO is $\log Q = n \log h$, or,

$$n = \frac{\log Q}{\log h},$$

but when $Q = 10$, the value of h will be the distance DA

or

$$h = 2.535.$$

Therefore,

$$n = \frac{\log 10}{\log 2.535} = \frac{1}{0.403} = 2.48.$$

Hence in general for the value $Q = 10$ for all lines passing through the origin n will equal $\frac{1}{\log h}$. Since angle α = angle α', the value of n for line BC will be the same as for line OA just found.

It will be seen that if the line is produced to cut DE, $Q = 10$, and

$$n = \frac{1}{\log h}$$

or the log of the intercept DA = the reciprocal of the exponent n (if the line is produced to cut RE as line QK, then we have

$$n = \frac{\log Q}{\log 10} = \log Q.$$

If now we make $h = 1$, then $\log h = 0$ and the equation

$$\log Q = n \log h + \log K$$

becomes

$$\log Q = \log K, \text{ or } Q = K,$$

and the value of K in the formula is the intercept on the Q-axis.

In Fig. 197, the line BC, prolonged, intercepts the Q-axis prolonged.

Instead of prolonging the Q-axis, we may find K by projecting point B vertically upward to G, then through G, draw GF parallel to BC and we find the intercept at $F = 2.9$. Since this is really the intercept of the Q-axis prolonged, which has values one-tenth those of OD, the intercept value K equals 0.29. This leads to the rule that for each transposition our values must be either multiplied or divided by 10, as the case may be.

This may be shown algebraically as follows:

Equation of line FG is

$$\log Q' = n \log h' + \log K',$$

n being constant since FG has been drawn parallel to BC, or

$$n \log h' = \log Q' - \log K'.$$

Also the equation of BC is

$$\log Q = n \log h + \log K,$$

or

$$n \log h = \log Q - \log K.$$

Dividing,

$$\frac{\log h'}{\log h} = \frac{\log Q' - \log K'}{\log Q - \log K},$$

but for points B and G, $h = h'$ and $Q = 1$, $Q' = 10$. Substituting,

$$1 = \frac{\log 10 - \log K'}{\log 1 - \log K},$$

$$-\log K = \log 10 - \log K',$$
$$\therefore K' = 10K,\ K = K' \div 10.$$

The values of h and Q given in Fig. 197 were too small to be plotted conveniently on log paper, so the values of h were multiplied by 10, and the values of Q by 60, to get them into conveniently plottable values. This line BC was then plotted from the values given for h' and Q'.

This multiplying of the values of h and Q has no effect on the slope of the line; or, in other words, it does not affect the values of the exponent n.

It makes no difference what we use as a multiplier for our values of either Q or h, the value of n remains constant; the line simply shifts parallel to itself along the h- or Q-axis.

Suppose the data had given a line HI. Then draw OK making the angle with the h-axis equal to the angle which HI would make with the axis of h; then n is equal to the log 2.95, but the sign is negative, which is indicated by the fact that the original line HI slopes from left to right downward.

$$\log 2.95 = 0.4698, \text{ or about } 0.47. \text{ Therefore } n = -0.47$$

When the line through O cuts the upper scale DE, as did the line OA, then n is greater than unity; when the line cuts the right-hand scale RE, n is less than unity.

To get K for this line HI, we have the intersection with the Q-axis at 4.5; therefore $K = 4.5$ and the equation is,

$$Q = 4.5h^{-0.47}, \text{ or } Q = \frac{4.5}{h^{0.47}}.$$

For the line LM, drawing ON making an equal angle with the h-axis, we find N intercepts at 5.4.

$$\log 5.4 = 0.732\ \frac{1}{0.732} = 1.39;$$

therefore, $n = -1.39$ (line slopes downward from left to right). Again to find K, drop L vertically downward to the h-axis; then from its intersection draw PS parallel to the original line LM. Thus we find S at 1.61, for this is $\frac{1}{10}$ the actual value of K since it was necessary to transpose from L to P in order to find the intersection S.

Therefore, the true value of K is 16.1, and again we have the equation of the line $LMQ = 16.1h^{-1.39}$.

If the top and the right-hand scales are divided into 10 equal divisions from 0 to 1, we have a method of obtaining the value of n without the use of a table of logarithms.

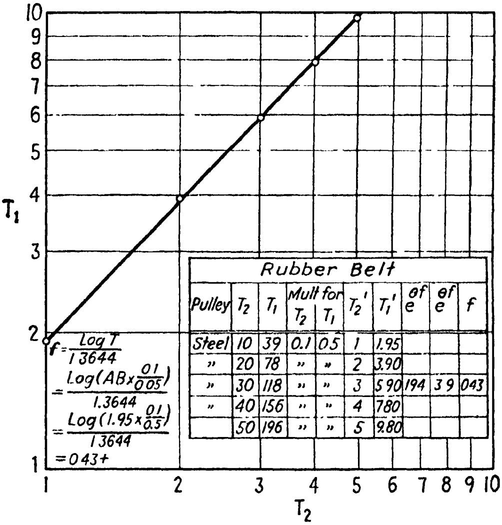
Whenever the line from the origin cuts the upper scale, as did OA, at A, then the reciprocal of the scale reading is the value of n sought and is always greater than 1. Thus OA cuts the upper scale at 0.405. Then.

$$n = \frac{1}{0.403} = 2.48.$$

When the line from the origin cuts the right-hand scale, as did line OK, at K, then the scale reading is the value sought and is always less than 1.

To determine the effect of the multipliers on the value of the intercept K.

Let $Q' = K'h'^n$ represent the equation of the line as actually plotted and $Q = Kh^n$ the equation of the line desired. The relationship of the lines being such that $h' = ah$ and $Q' = bQ$, *i.e.*, the values of h and Q were not plottable as values having their limits between 1 and 10, as demanded by the coordinates of the single bay logarithmic paper. These values had to be multiplied by such constants a and b as to make the new values plottable, *i.e.*, the equation of the line actually plotted is $Q' = Kh'^n$, while the line $Q = Kh^n$ is the desired line. As already proven, the value of n is the same for both lines, as its value is independent of the multipliers a and b.



Rubber Belt									
Pulley	T_2	T_1	Mult for T_2	Mult for T_1	T_2'	T_1'	$e^{\theta f}$	$e^{\theta f}$	f
Steel	10	39	0.1	0.5	1	1.95			
"	20	78	"	"	2	3.90			
"	30	118	"	"	3	5 90	194	3 9	043
"	40	156	"	"	4	780			
	50	196	"	"	5	9.80			

$$f = \frac{\text{Log } T}{1\,3644} = \frac{\text{Log}\left(AB \times \frac{01}{005}\right)}{1.3644} = \frac{\text{Log}\left(1.95 \times \frac{01}{0.5}\right)}{1\,3644} = 0\,43+$$

FIG. 198.—Logarithmic plot of typical test results for friction of rubber belt on steel pulley, Experiment 19.

To find the relationship between the K and K' values (K' being found from the actual line as plotted) proceed as follows: Since $h' = ah$, and $Q' = bQ$,

$$\log Q' = n \log h' + \log K',$$
$$\log Q' = \log b + \log Q, \text{ and } \log h' = \log a + \log K'.$$

Also from $Q' = K' h'^n$, we have

$$\log Q' = n \log h' + \log K'.$$

Therefore,

$$\log b + \log Q = n(\log a + \log h) + \log K',$$

or

$$\log K' = \log b + \log Q - n(\log a + \log h),$$

but

$$\log K = \log Q - n \log h, \text{ from } Q = Kh^n.$$

Subtracting,

$$\log K' - \log K = \log b - n \log a,$$

$$\log \frac{K'}{K} = \log \frac{b}{a^n},$$

or

$$K = \frac{K'a^n}{b}.$$

Therefore, knowing a and b and determining K' and n for the line actually plotted, the value of K may be found.

The analysis illustrated in Fig. 198, gives values as found for the rubber belt in Experiment 19.

The above analysis may be extended to the solution of equations of the form

$$Q = kh^n \pm Z,$$

in which Z is a constant.

PROPERTIES OF FUELS

Table XLVI.—Properties of Typical American Solid Fuels

Fuel	Percentage analysis as received								Higher heating value Btu per lb
	Moisture	Volatile matter	Fixed carbon	Ash	Sulphur	Hydrogen	Carbon	Oxygen	
Anthracite, stove	3.5	6.8	78.2	11.5	0.5	2.5	78.9	5.8	12,800
Pocahontas, lump	2.8	14.5	77.4	5.3	0.6	4.5	83.4	5.0	14,500
Pittsburgh, slack	3.4	31.7	56.5	8.4	1.0	5.1	74.4	9.7	13,600
Illinois, screenings	8.1	34.2	46.7	11.0	2.7	5.2	66.0	13.5	12,100
Coke, nut	4.5	1.5	84.5	9.5	1.0	...	....	....	12,400
Wyoming, subbituminous	15	36	42	7	...	...	....	....	10,000
Lignite	35	29	27	9	...	...	....	....	6,700
Peat (air dried)	25	48	21	6	...	...	....	....	7,000
Wood (oak, air dried)	17	....	....	0.5	...	5	40	....	5,500

TABLE XLVII.—PROPERTIES OF HYDROCARBON FUELS

Paraffin series, at 60°F, 14.7 lb per sq in. abs

Name	Chemical formula	Per cent of carbon	Lb air per lb fuel	Higher heating value, Btu per lb	Weight-density, lb per cu ft	Boiling point, °F
Methane	CH_4	75.00	17.28	23,700	0.042	−263
Ethane	C_2H_6	80.00	16.15	22,200	0.079	−130
Propane	C_3H_8	81.85	15.72	21,500	0.110	−40
Butane	C_4H_{10}	82.76	15.51	21,200	0.153	33
Pentane	C_5H_{12}	83.33	15.40	21,100	39.4	97
Hexane	C_6H_{14}	83.73	15.31	20,800	42.0	157
Heptane	C_7H_{16}	84.00	15.23	20,600	43.6	209
Octane	C_8H_{18}	84.21	15.16	20,600	44.5	256
Nonane	C_9H_{20}	84.37	15.12	20,500	45.5	300
Decane	$C_{10}H_{22}$	84.50	15.10	20,500	46.3	340

NOTES: Methane and ethane are the principal constituents of natural gas.

Propane and butane are sold as bottled fuel gases.

Octane is a comparison fuel for gasoline and has approximately the same ultimate analysis.

TABLE XLVIII.—EXHAUST-GAS ANALYSES BY VOLUME*

For petroleum fuels having a hydrogen-carbon ratio of 0.175

Lb air / Lb fuel	CO_2 %	O_2 %	CO %	H_2 %	N_2 %	Lb water / Lb fuel
11	8.76	0.15	9.14	4.66	77.08	0.972
12	10.18	0.44	6.65	3.39	79.13	1.043
13	11.60	0.59	4.31	2.20	81.09	1.122
14	13.02	0.62	2.10	1.07	82.97	1.205
15	13.23	1.35	0.99	0.50	83.72	1.247
16	12.62	2.49	0.68	0.35	83.65	1.256
17	12.00	3.55	0.48	0.25	83.51	1.261
18	11.45	4.49	0.30	0.16	83.39	1.267

* *N.A.C.A. Report* 616 (1937).

NOTE: These constituents total 99.79 per cent. The balance is largely methane. Both hydrogen and methane will be indicated as nitrogen by the usual Orsat analysis.

SPECIFICATIONS FOR PETROLEUM PRODUCTS

Oil Specifications.—Tables XLIX to LIV[1] give typical specifications and properties of American crude oils, fuel oils, lubricants, and gasoline. The name and description or classification of each of the crude and fuel oils are sufficient for identification, but further description is necessary to classify the lubricants. The government specifications recognize classes *A* to *D*

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[1] Compiled from *Bur. Mines, Tech. Paper* 323*A* and other sources.

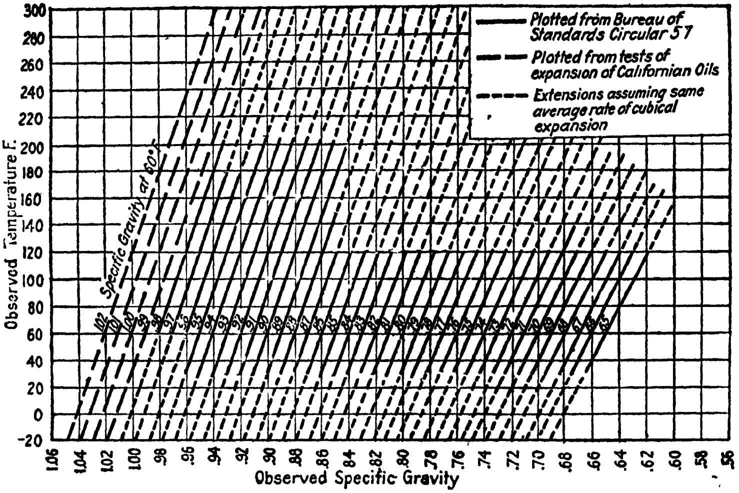
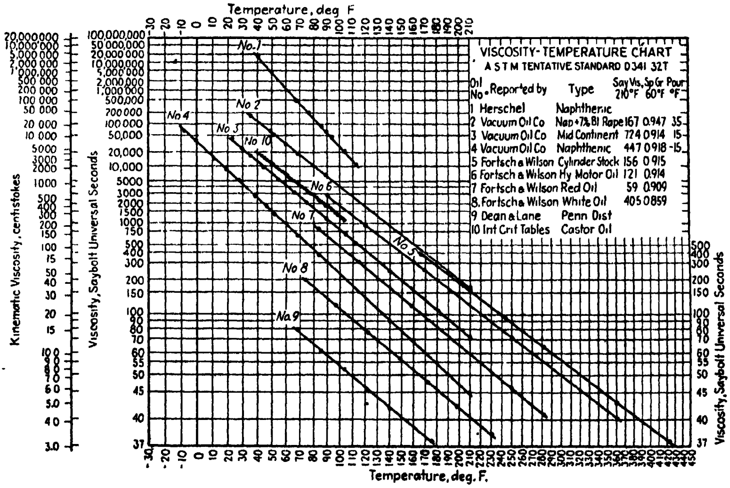


FIG. 199.—Chart for correcting specific gravity of petroleum oils for temperature, *i.e.*, for correcting observed specific gravity to specific gravity at 60 deg. fahr. (*Prepared by R. S. Danforth for Kinney Mfg. Co., Boston.*)



VISCOSITY-TEMPERATURE CHART

A S T M TENTATIVE STANDARD D341 32T

Oil No.	Reported by	Type	Say Vis. 210°F	Sp Gr 60°F	Pour °F
1	Herschel	Naphthenic			
2	Vacuum Oil Co	Nap + 7% Bl Rape	167	0.947	35
3	Vacuum Oil Co	Mid Continent	724	0.914	15
4	Vacuum Oil Co	Naphthenic	447	0.918	-15
5	Fortsch & Wilson	Cylinder Stock	156	0.915	
6	Fortsch & Wilson	Hy Motor Oil	121	0.914	
7	Fortsch & Wilson	Red Oil	59	0.909	
8.	Fortsch & Wilson	White Oil	405	0.859	
9	Dean & Lane	Penn Dist			
10	Int Crit Tables	Castor Oil			

FIG. 200.—A.S T.M. viscosity-temperature chart showing typical experimental curves. (Size of standard chart 16 × 21 in.)

inclusive as indicated in the table, though a numerical symbol system is also in use in the Navy. Class *A* oils are for general lubrication of engines and machinery where a highly refined oil is not required. Class *B* oils are used for the lubrication of turbines, dynamos, and high-speed steam engines using circulating and forced-feed systems. Class *C* covers only those oils used where service conditions necessitate an oil suitable for lubricating both turbines and internal-combustion engines. It is suitable for forced-feed systems. No fatty oils or compounds that are not derived from crude petroleum may be used in any of the above oils. Class *D* covers the grades of petroleum oil for lubrication of internal-combustion engines, except aircraft and diesel engines. The following remarks apply to the oils for designated uses:

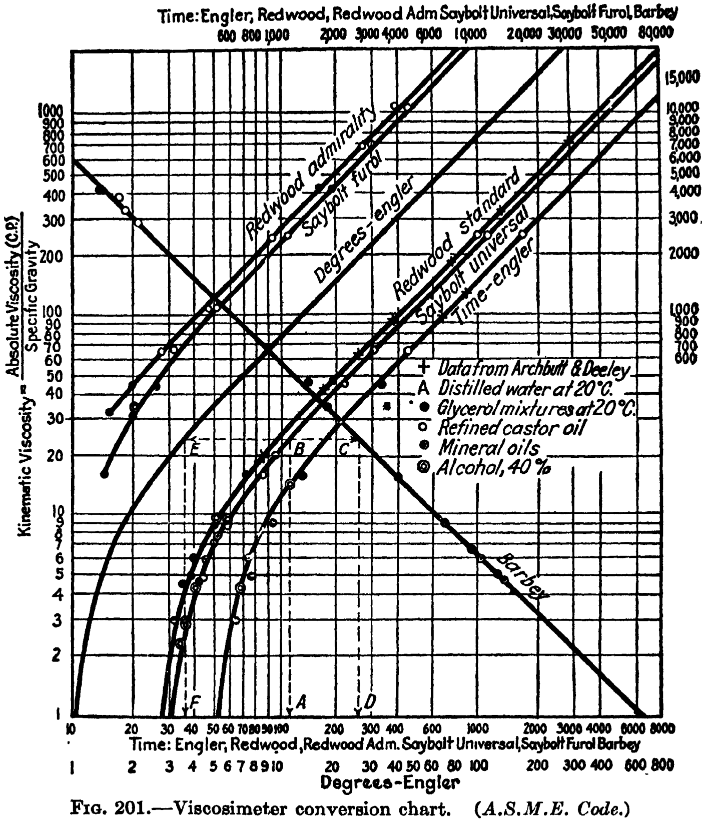


FIG. 201.—Viscosimeter conversion chart. (*A.S.M.E. Code.*)

Steam-engine Cylinders.—Steam cylinder *A* (compounded) covers the grades of well-refined petroleum oil used for lubrication of steam cylinders, noncondensing engines, and pumps where a compounded oil is required. Oil should be compounded with not less than 5 per cent or more than 7 per cent of acidless tallow or lard oil. Steam cylinder *B*, grades 1 and 2, are well-refined petroleum oils with no compounding.

Ice-machine Oils.—In addition to being used for the cylinders of ice machines, these oils are used for lubrication of pneumatic tools and for hydraulic systems. Only refined petroleum oils shall be used.

Air-compressor Oils.—These oils should be of high grade, highly refined, and give a high flash point and low carbon residue. Excessive quantities

TABLE XLIX.—PROPERTIES OF TYPICAL AMERICAN CRUDE OILS

Field	Physical properties			Chemical properties			
	Specific gravity	A.P.I. gravity at 60°F	Btu per lb	Carbon C	Hydrogen H	Oxygen O	Sulphur S
California	0.957	16.3	18500	86.58	11.61	0.74	0.82
Ohio	0.800	44.1	19580	83.40	14.70	1.30	0.60
Oklahoma	0.903	25.3	19650	87.93	11.37	0.19	0.41
Pennsylvania light	0.826	39.8	19930	82.00	14.80	2.20	1.00
Pennsylvania heavy	0.886	28.2	19210	84.90	13.70	1.40	
West Virginia	0.873	30.6	19800	84.3	14.1	1.6	
Texas	0.926	21.3	19400	83.26	12.41	3.83	0.50

TABLE L.—SPECIFICATIONS AND PROPERTIES OF LUBRICATING OILS

	Classes *A*, *B*, and *C* minimum values		Classes *A*, *B*, and *C* viscosity at 100°F maximum	Classes *A*, *B*, and *C*, pour point, maximum	Classes *C* and *D*, carbon residue, % maximum
	Flash	Fire			
Extra light	315	355	135 to 165	35	0.1
Light	325	365	180 to 220	35	0.2
Medium	335	380	270 to 330	40	0.45
Heavy	345	390	360 to 440	45	0.55
Extra heavy	355	400	450 to 550	50	0.70
Ultra heavy x	360	410	55 to 65	50	0.80
Steam cylinder *A*	475	...	120 to 150 at 210°	60	
Steam cylinder B. Grades (1) and (2)	475(1) 525(2)	... . .	135 to 165(1) at 210° 180 to 220(2)	60	4.5
Ice machine, grade 100	290	...	95 to 115		
Ice machine, grade 125	290	...	120 to 135	5	
Air compressor, single-stage	390	445	320 at 100° 50 at 210°	10	
Air compressor, multi-stage	400	450	410 at 100° 55 at 210°	10	
Turbine, direct-connected	......	...	150 to 300 at 100°		
Turbine, geared	......	...	300 to 700 at 100°		
Turbine, ring-oiled bearings	......	...	120 to 175 at 210°		

TABLE LI.—COMMERCIAL STANDARD FUEL-OIL SPECIFICATIONS CS 12-38*

Grade of oil	Flash point Min Max	Max water and sediment	Max pour point (1)	Distillation test	Max viscosity	Approx gravity	Calorific value per gallon
No. 1 (a distillate oil for use in burners requiring a volatile fuel)	110°F 165°F or legal	Trace	0°F†	Max 10% point 410°F Max. end point 560°F		38–40	135,000
No. 2 (a distillate oil for use in burners requiring a moderately volatile fuel)	110°F 190°F or legal	0.05%	10°F†	Max 10% point 440°F		34–36	138,000
No. 3 (a distillate oil for use in burners requiring a low-viscosity fuel)	110°F 230°F or legal	0.1%	20°F†		Saybolt universal at 100°F 45 sec	28–32	141,000
No. 4 (an oil for use in burners requiring low-viscosity fuel)	150°F	1.0%			Saybolt universal at 100°F 125 sec	24–28	144,000
No. 5 (an oil for use in burners permitting a medium-viscosity fuel) (Bunker B. Fed. Spec., 2d)	130°F or legal	1.0%			Saybolt-Furol at 122°F 40 sec	18–22	148,500
No. 6 (an oil for use in burners permitting a high-viscosity fuel) (Bunker C. Fed. Spec., 2d)	150°F	Water 1.75% Sediment 0.25%			Saybolt-Furol at 122°F 300 sec	14–16	152,000

* U. S. Department of Commerce, same as A.S.T.M. Tentative Specifications for Fuel Oils, D396-38T.

† Pour point may be specified whenever required by conditions of storage and use.

No. 3 or No. 4 oil usually specified as fuel for diesel engines.

of oil fed to an air compressor may collect in receivers and under certain conditions cause an explosion.

Steam-turbine Oils.—These oils should be highly refined, pure mineral oils with high demulsibility and resistance to oxidation. Ring-oiled bearings on small turbines are usually close to the steam casing and absorb much heat. Hence an oil of high viscosity is required.

TABLE LII.—TYPICAL SPECIFICATIONS FOR MOTOR OIL FOR LUBRICATION OF INTERNAL-COMBUSTION ENGINES

Grade S.A E *	10	20	30	40	50	60	70
Gravity °A.P.I.	29+	28+	27.5+	27+	26+	25.5+	25+
Flash °F . . .	405+	420+	430+	445+	460+	475+	500+
Viscosity:							
S.U.S. 0°F .	5,000– 10,000	10,000– 40,000					
S.U.S. 130°F	90–95	135–155	205–220				
S.U.S. 210°F				71–75	95-100	115-125	135-145
Viscosity index . . .	98+	95+	95+	95+	95+	95+	95+
Color, A.S.T.M.. .		5–	6–	7–	7–	8–	8–
Pour point not more than °F........	–5	0	10	15	20	20	25
Carbon residue, %. . .	0.15–	0.50–	0.70–	0.95–	1.25–	1.50–	1.80–
Corrosion.. .	Negative	Negative	Negative	Negative	Negative	Negative	Negative

The lubricating oils shall be refined petroleum oils without the admixture of fatty oils fatty acids, resins, soaps, or other nonhydrocarbon materials.

* For official S.A.E. viscosity specifications, see Table XIII, p. 171.

TABLE LIII.—GASOLINE SPECIFICATIONS

	Motor gasoline	Aviation gasoline, domestic grade
Distillation temperatures:		
10% evap, max....................	167	158
50% evap, max....................	284	212
90% evap, max....................	392	257
End point, max....................	437	311
Recovery, % min....................	95	97
Residue, % max....................	2	1.5
Loss, % max....................	......	1.5
Sulphur, % max....................	0.10	0.05
Gum, accelerated, mg max....................	......	6
Reid vapor pressure at 100°F, max........	12	7

The usual limitations (1940) for tetraethyl lead in aviation gasolines are 2 cu cm per gal for 80 octane and 4 cu cm gal for 90 or 95 octane.

When gasoline is purchased at retail, the two most common properties on which data are available are the gravity (see page 155) and the octane number (see page 458). Some states have set up gravity requirements (say, 58 to 60° A.P.I. for "regular" motor grade), but usually the gravity is only a rough index. The companies that supply the antiknock compounds check the analyses of the gasolines in which these compounds are used.

TABLE LIV.—COEFFICIENT OF EXPANSION OF PETROLEUM OILS

Group	Range of gravity, A.P.I.	Coefficient of expansion
1	.. to 34.9	0.0004
2	35 to 50.9	0.0005
3	51 to 63.9	0.0006
4	64 to 78.9	0.0007
5	79 to 88.9	0.0008
6	89 to	0.00085

FLUID-FRICTION FACTORS

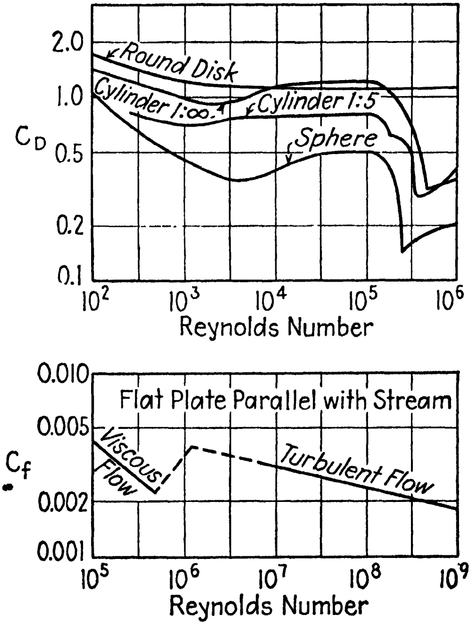


FIG. 202.—Coefficients of friction-drag of bodies in an air stream. Coefficients apply to Eq. 50, p. 262. C_d applies to geometrical shapes, when projected area is used. C_f is for a flat plate parallel to the stream, when total area is used.

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